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POWER PLANT TESTING

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POWER PLANT TESTING

A Manual of Testing Steam Generating Equipment, Engines, Turbines, Pumps, Refrigerating Machinery, Fans, Fuels, Lubricants, Etc.

BY

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FOURTH EDITION
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PREFACE TO THE FOURTH EDITION

POWER plant testing, since the last revision of this book, has become increasingly exact. Chiefly responsible for this change has been the introduction of high steam pressures and of rotary prime movers in large units of plant equipment. Sheer economic necessity demanded refinements in testing.

To meet this need, new methods and refinements of old methods have been developed which have resulted in new and higher requirements for all standard power plant testing. Most of the changes were based upon the recommendations of the national engineering societies.

Especially in temperature and pressure measurements, more consistent results are being obtained today by the new approved methods than were ever possible by the old practices. Then again, improved methods are responsible for standardizing the testing of lubricants and fuels, a very great advance.

The Power Test Committee of the American Society of Mechanical Engineers has practically completed its revision of the testing codes and, with the approval of the Council of the Society, has put them now into relatively permanent form. New codes and testing methods, usually somewhat abbreviated, have been incorporated in this book. Included in the chapter on Flow of Fluids is the code proposed and recommended by the A.S.M.E. Research Committee to determine for "guarantee" tests the steam consumption of noncondensing and bleeder steam turbines by the calculated flow through suitable nozzles. This recommendation is important because it puts American testing methods in better agreement with the European than before. To bring the book strictly up to date in this way has involved the rewriting of large sections.

Many engineers and teachers have contributed to this revision. Special mention should be made of the contributions of the following professors and engineers to this edition and to the one preceding: L. S. Marks, Harvard School of Engineering; Joseph H. Keenan, Stevens Institute of Technology; J. J. Eames and Dean Fales, Massachusetts Institute of Technology; J. E. Emswiler and C. H. Fessenden, University of Michigan; R. L. Daugherty, California Institute of Technology; R. S. King, Georgia Institute of Technology; F. W. Marquis, Ohio State University; Harry L. Parr, Columbia University; W. Trinks, Carnegie Institute of Technology; Herman Diederichs, Sibley School of Engineering, Cornell University; N. C. Miller, Rutgers University; Charles F. Shoop, University of Minnesota; R. U. Fittz, Tufts College; F. H. Vose, Case School of

Applied Science; the late A. J. Wood, Pennsylvania State College; G. S. Wilson, University of Washington; the late E. H. Lockwood, Yale University; J. P. Calderwood, Kansas State College; A. A. Potter, President, American Society of Mechanical Engineers; S. A. Moss, Thomson Research Laboratory, General Electric Company; Francis Hodgkinson, Westinghouse Electric and Manufacturing Company; John York, Stone and Webster Engineering Corporation; R. E. Sprenkle, Bailey Meter Company; G. A. Stetson, Editor, *Mechanical Engineering*, and J. F. Wostrel, Massachusetts Department of Education.

The author is always glad to answer correspondence with teachers and engineers relating to questions that arise in testing work but that cannot be taken up in detail in a book of moderate size.

Information regarding typographical errors in any part of the book, but especially in the Testing Codes, and suggestions in regard to subject matter that will add to the usefulness of the book to engineers and students will be gladly received.

An unusually complete index has been prepared for this edition with the intention of making a *reference handbook* on the testing of prime movers and other mechanical equipment of power plants.

JAMES AMBROSE MOYER.

STATE HOUSE, BOSTON, MASS.,
May, 1934.

PREFACE TO THE FIRST EDITION

IN the preparation of this book the object in view has been primarily to give in a small volume, somewhat in detail, the generally approved methods of testing engines, turbines, boilers, and the auxiliary machinery usually found in power plants, as well as to present more or less complete descriptions of the various kinds of apparatus used and the calibrations required for accurate testing. In addition to this subject matter, chapters have been prepared on the testing of fuels, refrigerating and hydraulic machinery, as well as centrifugal fans, air compressors, and lubricants.

As a book for students in laboratory courses it is intended particularly for use in large classes in which at the beginning of the laboratory periods it is necessary to begin at the same time a number of different experiments and tests. On this account care has been taken to state as clearly as possible the descriptions of the apparatus to be used and the precautions to be observed to secure accuracy in the results. Students should be expected, however, to rely to some extent on their own initiative.

In most respects the book is probably complete enough in descriptive matter and in general instructions so that very little lecture-room work is needed for at least elementary courses. It is the author's opinion that students in experimental engineering laboratories should not receive a great deal of assistance in planning and conducting tests. Sometime they must learn to be resourceful and independent of the "school" type of instruction, and obviously the sooner this is appreciated by both instructors and students the greater will be the benefits. At least for very small classes the better plan is the one advocated years ago by a famous educator, that students working in laboratories when assigned the work of testing a machine, a new type of airplane engine, for example, should have very simple instructions such as: "Make tests of this new type of engine, find out what you can about it and report your results." It is to be hoped that the particular method of teaching in laboratories, known familiarly as "feeding with a spoon," has disappeared in present-day instruction in technical schools and colleges.

Quite a large part of the training required for one to become accurate and reliable in the work of observing and interpreting the results of tests of machinery consists in becoming familiar with the details of the adjustment and calibration of the various instruments, so that they may be used intelligently.

Although in the arrangement of the chapters the use of the book by students was given the most careful consideration, yet as a whole the

needs of the "practical" man were not lost sight of, and it is hoped that the author's experience when working with this group of readers in testing both large and small power plants has helped to make the book interesting and helpful to them. The book is intended to be also a manual giving useful information in a more or less limited way to those professional engineers having the advantages of a technical training, but who are not thoroughly familiar with the most up-to-date methods of testing.

In many cases not nearly all the results that should be calculated to make up a complete report are mentioned. It is the opinion of the author that in a textbook it is desirable that no more than very general instruction should be given regarding the conduct of the test, the quantities to be calculated, and the form and tabulations expected in a report. Such details should be left in the hands of the instructor. Because the size of the book was limited it was necessary to omit explanations of the methods of calculating many interesting and more or less applicable results from tests. In the more extended courses it is believed that the instructors can readily fill in these omissions.

The author is particularly indebted in the preparation of this book to Dean M. E. Cooley and Professor J. R. Allen, of the University of Michigan; Professor L. S. Marks, of Harvard University; Professor H. W. Spangler, of the University of Pennsylvania; Dean W. F. M. Goss, of the University of Illinois; Professor C. H. Peabody, of the Massachusetts Institute of Technology; Professor L. V. Ludy, of Purdue University; Professor A. M. Greene, of Rensselaer Polytechnic Institute; Professor C. C. Lorentzen, of New York University; Professor E. J. Fermier, of the Mechanical and Agricultural College of Texas; Professor E. A. Fessenden, of the University of Missouri; Professor F. H. Sibley, of the University of Alabama; Dr. C. P. Steinmetz and Mr. Richard H. Rice, of the General Electric Company; Mr. H. R. Kent, Vice-president, Westinghouse, Church, Kerr & Company; Mr. J. R. Bibbins, of the Arnold Company; Mr. R. A. Smart, of the Westinghouse Machine Company; Mr. St. John Chilton, of the Allis-Chalmers Company; and Mr. G. E. Wallis, New York City.

JAMES AMBROSE MOYER.

ANN ARBOR, MICH.,
August, 1911.

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POWER PLANT TESTING

CHAPTER I

MEASUREMENT OF PRESSURE

The simplest instrument used for measuring pressure is a glass tube bent into the shape of the letter U, as illustrated in Fig. 1. When such a tube, called a **manometer** or **U-tube**, is partly filled with a liquid, usually water or mercury, and is connected as at *A* in the figure by means of tubing to the container in which the pressure is desired, there will be observed a difference in the level of the liquid corresponding to the pressure. If the end of the tube at *B* is open to the

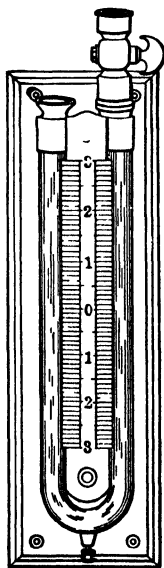


FIG. 2.—Manometer or U-tube with graduated scale.

atmosphere, then the difference in the level of the liquid in the two branches of the U-tube called “legs” measured in inches, multiplied by the weight of a cubic inch of the liquid in pounds, gives the *difference* in pressure in *pounds per square inch* between that in the container and atmospheric pressure. When the level in the leg *B* is higher than in *A*, then the pressure measured is greater than atmospheric and is called **gauge pressure**¹ to distinguish it from the other condition when the level in the leg *A* is higher than in *B*, that is, when the pressure is less than atmospheric. In the latter case we speak of **vacuum** or negative pressure.

As such instruments are usually constructed, a scale suitably graduated for measuring the difference between the levels of the liquid in the tube is placed between the two legs, as shown in Fig. 2. Still another type is illustrated in Fig. 3. In a manometer of this kind one leg can be made very short if it is correspondingly large in diameter. If the short leg of a manometer is made large



FIG. 1.—A U-tube for measuring pressures.

¹ The ordinary pressure gage (see pp. 14 and 17) indicates how much the pressure that is being measured is above the atmospheric pressure. The pressure as shown by a gage (gage pressure) added to atmospheric pressure is called total or **absolute pressure**.

enough in diameter to have a practically invariable level, and the scale is set with its zero at the level in the short leg, then the level in the long leg will indicate directly inches of pressure or of vacuum, as the case may be. A commercially made vacuum gage of this kind is shown in Fig. 4. The end of the tube corresponding to the short leg in Fig. 3 is shown at *A*. When manometers are to be used for pressure or vacuum measurements of **steam**, a condensation trap (*B*, Fig.

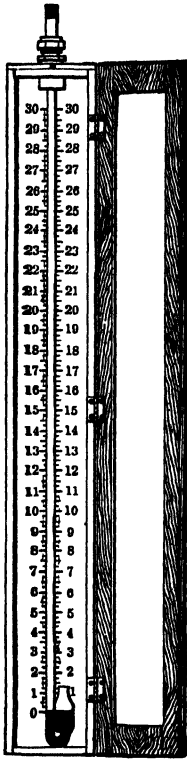


FIG. 3.

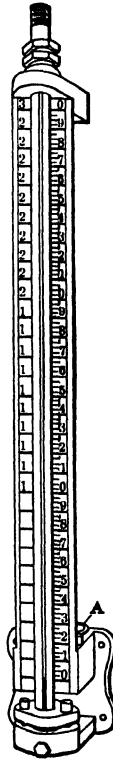


FIG. 4.

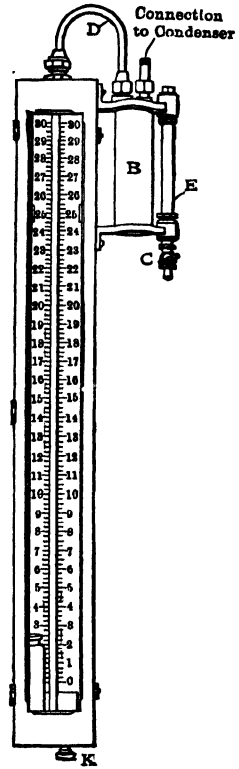


FIG. 5.

Typical mercury vacuum gages.

5) is often employed to prevent the passage of steam into the glass tube, in which it would form a water column on the top of the mercury, for which a correction¹ would have to be made. To be effective, the condensation trap *B* must always be partly filled with water. It must not, however, be allowed to become completely filled so as to discharge water through the pipe *D* joining the trap with the glass vacuum tube. A glass

¹ Correction for water on the top of a mercury column is most conveniently made by dividing the length of the water column by the specific gravity of mercury (13.6) and adding this equivalent length to the mercury column on which the water rests.

tube *E*, called a **water gage**, shows the level of the water in the trap and should always be kept clean. Cock *C* is provided for draining off excess of condensation.

The graduated scales on vacuum gages, like Figs. 3, 4, and 5, must be adjustable so that the zero can be made to coincide exactly with the level of the mercury when both legs are open to atmospheric pressure. Scales are usually arranged to be raised and lowered by turning a milled knob, like *K* in Fig. 5, which is connected by a screw thread to the scale. Vacuum gages, like Fig. 3, without ready means for adjustment cannot be arranged to indicate accurately vacuums that are varying, because

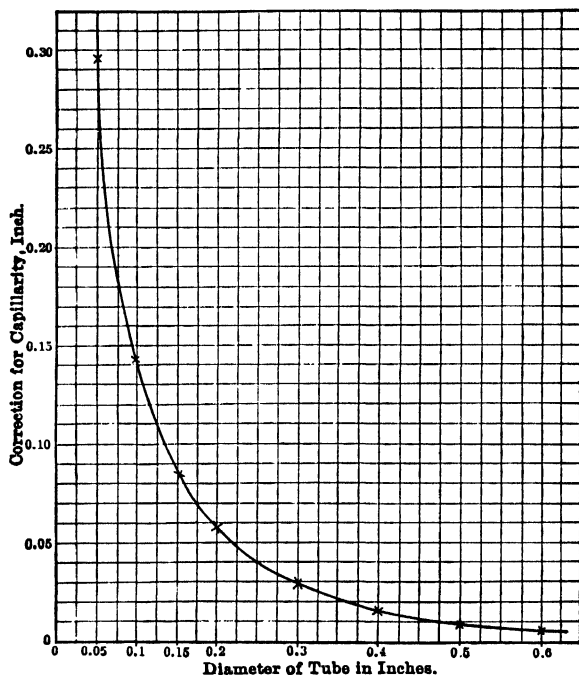


FIG. 6.—Curve of capillarity corrections for mercury columns.

the measure of the vacuum is the difference in level between the columns in the two legs. In all vacuum manometers, unless the area of the reservoir is very large, either the scale must be shifted or the level in the reservoir changed for varying magnitudes of vacuum.

Manometers or U-tubes of very small diameter when filled with mercury may be affected by capillarity to such an extent that, in order to obtain the true height corresponding to the pressure, a correction must be added. It is not at all unusual to find manometers used for vacuum gages to be comparatively small in diameter, and, unless the graduations of the scale have been corrected for the error due to capillarity, the proper allowances must be made for all observations. Figure 6 shows by a curve

the values of this correction as determined by Pullen for mercury columns of various diameters.

Mercury columns should be read at the top of the meniscus and water columns should be read at the bottom. In this way, except in very small tubes, the errors due to capillarity may be regarded as negligible. If the graduated scale is made to lap over nearly half the width of the tube on each side, as illustrated in Fig. 2, and all the observations are taken at the meniscus in the middle of the tube, a remarkable degree of accuracy is obtainable.

Correcting the Readings of Mercury Columns.—For calculations relating to the Power Test Codes of the American Society of Mechanical Engineers (A.S.M.E.), the height of all mercury columns must be corrected to the value which would be obtained if each mercury column were at the temperature of 32°F.

The height of any mercury column is to be measured by a scale reading standard *inches*. The correction for the temperature of the scale is small and therefore is not always necessary in engineering work.

TABLE I.—TEMPERATURE CORRECTIONS FOR BAROMETERS AND MERCURY COLUMNS

Temperature of column, degrees Fahrenheit	Observed reading of column, inches of mercury								
	16	18	20	22	24	26	28	30	32
Add									
—20	0.07	0.08	0.09	0.10	0.11	0.11	0.12	0.13	0.14
—10	0.06	0.06	0.07	0.08	0.08	0.09	0.10	0.11	0.11
0	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08
10	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05
20	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.02
30	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Subtract									
35	0.01	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.02
40	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03
45	0.02	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05
50	0.03	0.03	0.04	0.04	0.05	0.05	0.05	0.06	0.06
55	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08
60	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09
65	0.05	0.06	0.07	0.07	0.08	0.09	0.09	0.10	0.10
70	0.06	0.07	0.07	0.08	0.09	0.10	0.10	0.11	0.12
75	0.07	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.13
80	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15
85	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16
90	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.17	0.18
95	0.10	0.11	0.12	0.13	0.14	0.16	0.17	0.18	0.19
100	0.10	0.12	0.13	0.14	0.15	0.17	0.18	0.19	0.20

Table I for temperature correction, which was prepared by the U. S. Weather Bureau, is "precise and includes a slight correction for temperature of brass barometer scales." The reading of a barometer is sometimes given to the nearest thousandth of an inch. It is stated, however, in the Test Codes of the A.S.M.E. that the "nearest hundredth of an inch will be satisfactory for usual engineering purposes."

Mercury barometers are instruments designed to measure atmospheric pressure accurately. The one shown in Fig. 7 consists of a glass tube, closed at the top and enclosed in a metal casing *T*. The open end of the glass tube at the bottom dips into a cup *C* so that it is well below the level of the mercury, with which the cup is partly filled. The cup has glass sides and is closed at the bottom by a leather bag resting on a movable disk. By means of the screw *S* the level of the mercury can be raised or lowered until the surface of its meniscus just touches the tip of a fixed ivory point *P* shown in Fig. 7a. Atmospheric pressure acts

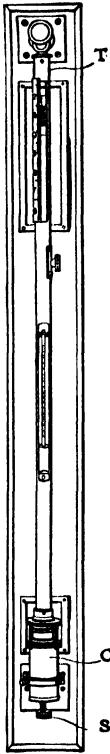


FIG. 7. — Mercury barometer.

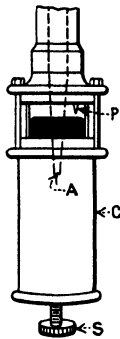


FIG. 7a. — Detail of cup in mercury barometer.

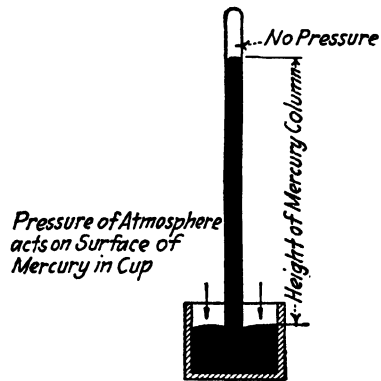


FIG. 7b. — Principle of barometer.

upon the mercury in the cup so that the height of the mercury column in the tube which is thoroughly exhausted of air is a measure of this pressure. A vernier is generally provided alongside of the scale to assist in accurate observations of the height, which is measured to the top of the meniscus. For every observation the level of the mercury in the cup *C* must be adjusted accurately to the tip of the ivory point. The equipment of a modern power plant is incomplete without a reliable and accurate barometer. The principle of operation of this instrument is illustrated in Fig. 7b.

Appreciable pressure differences are sometimes created in the vicinity of blowers supplying air to boilers, turbine generators, and similar

equipment. Such pressure differences may influence the reading of a barometer located nearby. In cases of this kind, the barometer must be located close to the mercury vacuum gages on a condenser. Similarly, in cases where the barometer and mercury vacuum gages are not at the same level, correction must be made for the difference in level. The number of feet measuring this difference in elevation of the barometer and the gages is measured "between the center line of the apparatus under test" and the barometer *cistern* level. There is also a correction in some cases for the elevation difference of mercury U-tubes (see Table II, page 7).

A mercurial barometer is not a portable instrument, so that it should be installed in a permanent place. In usual cases, the barometer should be read three times for a test: (1) before, (2) after, and (3) once during the period that the test is in progress.¹

Standard barometric pressure for comparison is now usually taken as 29.92 inches of mercury at a temperature of 32°F. The recommended correction is made by using Table I. Corrections are made for both the cubical expansion of mercury (coefficient = 0.000100) and for the linear expansion of the scale (coefficient of brass = 0.000011). If H is the barometric reading in inches at $t^\circ\text{F.}$, then the equivalent height H_0 corrected for temperature to 32°F. is

$$H_0 = H \left\{ 1 - \frac{(0.000100 - 0.000011)(t - 32)}{1 + 0.000100(t - 32)} \right\}.$$

This equation is equivalent in English units to the standard correction adopted by the International Bureau of Weights and Measures.²

Some observers, however, have adopted 62°F. instead of 32°F. as the standard for the **brass scale**, making the correction considerably more complicated. Whether the scale is corrected to 32 or 62°F. actually changes the results very little.

Vacuum measured with a mercury manometer must be corrected to standard temperature conditions in the same way as barometer readings are corrected. Frequently wooden scales are used on vacuum manometers. If the scale has been cut so that the length is along the grain, the expansion is very small (see Table IV in the Appendix) and the equation above can be simplified for most practical purposes to

$$H_0 = H \{ 1 - 0.0001 (t - 32) \}$$

when 32°F. is the standard for comparison.

It often happens that there are tens and even hundreds of feet difference in elevation between the location of the barometer and that of the

¹ According to A.S.M.E. Test Codes, it is permissible, in tests where no special accuracy is required, to obtain barometer readings only twice during a test, once before and once after.

² BROCH, *Bull. Intern. Bur. Weights and Measures*, 1887.

vacuum gages. It then becomes necessary to correct the reading of the vacuum gages to the **elevation of the barometer**. The correction is about 0.11 inch¹ per 100 feet for elevations less than 1,000 feet for normal power-plant temperatures. For high elevations and extreme temperatures see Table II. This altitude correction is to be added to the vacuum reading when the vacuum gage is elevated above the barometer, and *vice versa*. This correction for elevation is often serviceable for determining the approximate barometer reading at a place where no barometer is available, by applying the corrections to the barometer observation at the nearest station of the U. S. Weather Bureau. Observations reported by the Weather Bureau are at sea level and 32°F. To use these data it is therefore necessary to know the elevation above sea level of the place where the test is being made, and all observations made with mercury columns must be *reduced to the equivalent at 32°F.* to obtain *correct absolute pressures* (see Report of Power Test Committee of A.S.M.E. in *Journal of A.S.M.E.*, November, 1912, page 1696).

TABLE II.—CORRECTIONS FOR BAROMETERS AND PRESSURE GAGES, IN INCHES OF MERCURY FOR DECREASE OF ATMOSPHERIC PRESSURE PER 100-FOOT INCREASE IN ELEVATION

Mean altitude, feet	Mean atmospheric temperature, degrees Fahrenheit						
	-20	0	20	40	60	80	100
0	0.13	0.12	0.12	0.11	0.11	0.10	0.10
1,000	0.12	0.12	0.11	0.11	0.10	0.10	0.10
2,000	0.12	0.11	0.11	0.10	0.10	0.10	0.09
3,000	0.11	0.11	0.10	0.10	0.10	0.09	0.09
4,000	0.11	0.10	0.10	0.10	0.09	0.08	0.08
5,000	0.10	0.10	0.10	0.09	0.09	0.08	0.08
6,000	0.10	0.10	0.09	0.09	0.08	0.08	0.08
7,000	0.10	0.09	0.09	0.09	0.08	0.08	0.08

Obviously, temperature and elevation corrections are avoided when the barometer and the vacuum manometer are hung very near each other.

Precautions in the Reading of Mercury Barometers.—The Power Test Committee of the A.S.M.E. has listed the following precautions to be observed when taking atmospheric pressure observations with a mercury barometer:

1. The thumbscrew at the bottom of the mercury cup must be adjusted so that the top of the mercury just touches the ivory point (*P* in Fig. 7a), and then the slider, carrying the vernier, must be adjusted to the extreme top of the mercury-column meniscus.

¹ A cubic foot of air at average atmospheric temperatures weighs about 0.078 pound or $(0.078 \times 100) \div 144 = 0.0542$ pound per square inch for 100 feet of elevation, which is equivalent to $0.0542 \div 0.4912$ or 0.11 inch of mercury per 100 feet.

2. Both of these adjustments are best made by placing a piece of white paper behind the barometer, and making the adjustment so that the thread of white above the mercury first has an appreciable width and is then decreased until it just disappears. The ivory point must not make a dimple in the mercury. If the mercury in the cup is bright enough, the ivory point and its image may be seen to coincide. In making either adjustment, if the screw has been turned too far, go back enough to show a visible white space, and again start decreasing this space until it disappears.

3. The eye of the observer must always be on the level of the mercury being observed.

4. The tube and the cup must be tapped lightly while making the adjustments and taking the reading.

5. The reading of the vernier must then be found and recorded. Care must be taken to record the correct decimal corresponding to the last vernier division. This gives the *uncorrected barometric pressure*. For accurate work, reset the ivory point on the mercury level and take a check reading.

EXAMPLE OF A MERCURY-BAROMETER READING, CORRECTION, AND REDUCTION (AT RIVER WORKS, MASS., GENERAL ELECTRIC COMPANY)

	Inches Mercury
Uncorrected barometric pressure (actual reading).....	29.80
Barometer temperature, 75°F.	
Temperature correction (Table I).....	−0.13
Calibration correction (must always be made).....	−0.01 *
Barometric pressure (at barometer elevation).....	29.66
Level of barometer cistern <i>below</i> turbine center line, 16 ft.	
Elevation correction (Table II).....	−0.02
Barometric pressure at elevation of turbine center line.....	29.64
Reduced reading $29.64 \times 0.4912 = 14.56$ lb. per sq. in. =	
absolute barometric pressure at elevation of turbine center line	

* The original data included a *gravity correction* of 0.01 inch, which was made to reduce the barometer reading to the value which would occur if gravity (at the location of the barometer) had the international standard value of 32.1740 feet per second per second." A tabulation of values of this correction is given in "Instruments and Apparatus," A.S.M.E. Test Codes, Part II, Chap. VI, p. 16.

Calibration of Barometers.—According to the specifications of the Test Codes of the A.S.M.E., mercury barometers are to be calibrated by comparison with U. S. Weather Bureau readings. It is usually desirable to make several comparisons on successive days and use the mean calibration correction.

A mercury barometer is not a portable instrument and should not be moved or disturbed during use or calibration and "usually should not be taken to a weather bureau station for calibration." The calibration

of a mercury barometer *shall* be made during the course of any "set of tests" made under the Power Test Codes.

The difference between the reading of a barometer corrected for temperature and gravity (elevation) by Tables I and II, where tests are made and the reading for the same location as obtained from the U. S. Weather Bureau information, with the proper corrections for time, distance, and elevation, gives the *calibration correction* which must be applied to each reading of the barometer.

Possible Errors in Mercurial Barometers.—In the case of mercurial barometers the error is usually due to one or more of three causes: (1) an error in the scale, (2) an error in the scale zero, and (3) the presence of air in the glass tube above the mercury. Air in the glass tube or an error in the scale will give a variable error, while an error in the scale zero will give a constant error. However, the reading of the barometer usually varies so little during a period of test that the error due to any of these causes may, for engineering purposes, be taken as a constant error.

EXAMPLE OF CALIBRATION OF A MERCURY BAROMETER

Uncorrected barometric pressure.....	29.800 in. mercury
Check reading.....	29.801 in. mercury
Temperature.....	75°F.
Height of mercury cistern above floor.....	3 ft.
Floor elevation above mean sea level.....	16.82 ft.
Mean uncorrected reading.....	29.80 in. mercury
Temperature correction (Table I).....	-0.13 in. mercury
Elevation correction, 20 ft. at 75°F. (Table II)...	+0.02 in. mercury
Corrected reading, reduced to sea level.....	29.69 in. mercury
Weather-map reading, reported by Boston station of U. S. Weather Bureau 8:00 A.M. Eastern Standard Time, June —, —, reduced to sea level.....	
	29.68 in. mercury
Calibration correction.....	-0.01 in. mercury

Standard Barometric Pressure for Condensers, Engines, and Turbines.—When a rated vacuum or rated back pressure is given, it is to be understood, if not otherwise specified, that the *standard* barometric pressure is 29.92 inches¹ of mercury at 32°F., for a locality at sea level.

¹ Another standard barometric pressure, which has been described in some textbooks and steam charts, is based upon the assumption that, when the term "30-inch barometer" is used, the standard atmospheric pressure of 29.92 inches of mercury column at 32°F. is meant. Accordingly, on this basis the given value of 30 inches is understood to be 30 inches at some particular condition of temperature, which when corrected to 32°F. will give a 29.92-inch mercury column. This temperature has been worked out and found to be about 59°F., and systems have therefore been used in which all mercury column heights have been reduced to values corresponding to a temperature of about 58°F.

Table I (page 4) is used in correcting the readings of a barometer, mercury gage, or mercury U-tube of any kind. The correction given reduces the reading to the value which would be obtained if the mercury column were at the standard temperature of 32°F. The table also includes a slight correction for the temperature expansion of a brass scale, which is taken as being correct at 62°F. The scale-temperature correction is usually negligible for engineering purposes. To correct observations of vacuum to equivalent vacuum compared with (or "referred to") 29.92-inch barometer, the difference between the corrected barometer and 29.92 inches is *added* to the observed vacuum when the barometer is *less* than 29.92 inches and is *subtracted* when *greater* than 29.92 inches. This is the method generally used.

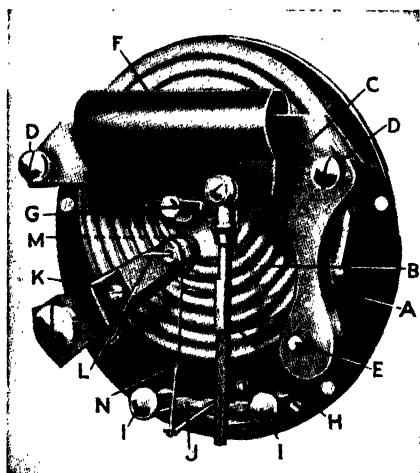


Fig. 8.—Diaphragm mechanism of aneroid barometer.

Aneroid Barometer.—Atmospheric pressure is sometimes measured by a mechanical device called an aneroid. It is simply a delicate pressure gage which is more conveniently portable than a mercury barometer.

It depends in principle for its operation on the expansion of a thin corrugated metal disk which responds elastically to a difference of pressure on its face. Figure 8 shows the mechanical details of an aneroid. Two corrugated disks are joined together at their edges to form a vacuum box *B* thoroughly exhausted of air, which is attached to the base plate *A* of the instrument. A so-called "bridge" *C* spans the vacuum box *B*. An adjusting screw *E* is provided for setting the aneroid to read correctly for a known atmospheric pressure by raising or lowering the bridge *C*, thus slightly adjusting the vacuum box *B* and therefore the reading of the aneroid. A steel spring *F* slides in the back part of the bridge *C* and is attached to the vacuum box *B* by means of the knife edge *G*. The steel spring *F* functions to oppose the tendency of the vacuum box *B* to collapse. This spring is secured to the base plate so that the upper disk of *B* will move upward or downward with the spring *F* according to changes of atmospheric pressure. A slender arm *H* is attached to the spring *F* so as to give a magnified movement corresponding to the upward or downward movement of the vacuum box *B*. An increase in atmospheric pressure causes the arm *H* to move downward and thus rotates the regulator *J* outward from the mechanism. Another arm is set

in an upward direction from the regulator *J* which, at its upper end, magnifies the movement to be indicated by the instrument. The pin *L* passing through the end of the lever *K* supports the indicating needle (not shown in the figure). A hair spring *M* is fitted to the pin *L* to keep taut the light chain *H* connecting the regulator *J* to the pin *L*.

An aneroid barometer should include a device which will compensate for temperature changes. If a proper calibration is made, it is permissible to use this instrument for work in connection with A.S.M.E. Power Test Codes, "where accuracy in the determination of barometric pressure is not of primary importance." At best, however, the temperature correction of this kind of instrument is uncertain. Sometimes there is appreciable hysteresis (page 27).

The A.S.M.E. Test Codes recommend that a mercury barometer should always be used. In many cases, pressures less than atmospheric are given in inches of mercury, vacuum. In such cases, the vacuum recorded is that read on the scale of a mercury column. Such vacuum readings must be corrected in the same way as barometer readings for temperature according to Table I, before subtracting this reading from the barometric pressure to obtain the absolute pressure. Such absolute pressures in *inches of mercury* must always be corrected to 32°F. Most engineers place more reliance on aneroids than their accuracy permits.

Precautions in the reading of aneroid barometers are also given as follows:

1. The instrument must be tapped lightly while taking a reading.
2. The eye must be held directly on a line at right angles to the tip of the needle point, to avoid parallax.
3. Interpolation between the graduations must be made by estimation to give the last figure in the reading.
4. The instrument must be read in the same position and preferably in the same location as when calibrated.

Recording Barometer or Barograph.—The mechanism of an aneroid barometer is designed so that, if the pointer has a pen attached to its free end, which will move by clockwork over a sheet of paper revolving on a suitable drum, a continuous barometric record is obtained. This instrument is, of course, subject to all the inaccuracies of the ordinary aneroid barometer, plus those due to lost motion of the pen mechanism and pen friction. For direct barometric readings, this instrument is not recommended. It may be satisfactorily used, however, in testing work in some cases to show variations of barometric pressure during a test, often in conjunction with "good" barometric readings, obtained by one of the approved methods. In this connection, however, it should be observed that hysteresis in the instrument as in the aneroid may be a disturbing time element.

TABLE III.—RANGES AND PERMISSIBLE ERRORS OF BAROMETERS

Type of barometer	Range of scale, inches of mercury	Permissible error, inches of mercury
Mercury barometer.....	25 to 32	0.001 to 0.002
Aneroid barometer.....	26 to 32	0.01 to 0.05
Recording barometer.....	26 to 32	0.10 to 0.20
Weather Bureau information ¹	20 to 32	0.01 to 0.10

¹ Barometric pressure obtained by telephone or correspondence for an exact time, and carefully corrected for difference in elevation and distance, will give results accurate to a hundredth of an inch. Uncertainties as to difference in elevation, time, or distance may make errors of a tenth of an inch or more.

Range and Accuracy of Barometers.—Mercury and aneroid barometers have usually a range of from 25 or 26 to 32 inches of mercury. Special barometers for use in mountainous regions begin at 15 or 20 inches. Table III gives the range of scale in inches of mercury and permissible error as a fraction of an inch of mercury for the various types of barometers.

Absolute Pressure Gages.—Simplest forms of instruments for indicating directly absolute pressure consist of a mercury barometer and a vacuum gage placed side by side, with a sliding scale between them, as illustrated in Fig. 9. When the zero is adjusted so as to be exactly opposite the top of the mercury

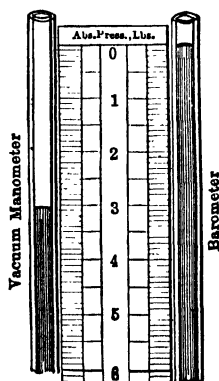


FIG. 9.—Simplest form of absolute pressure gage.

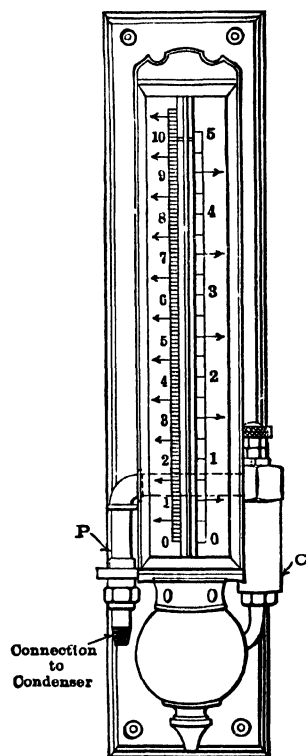


FIG. 9a.—Commercial form of absolute pressure gage.

column in the barometer, the reading on the scale opposite the level in the vacuum gage is the absolute pressure. Obviously, the scale can be graduated to indicate inches of mercury, pounds per square inch, or any similar units of absolute pressure. A simpler commer-

cial form of absolute pressure gage is shown in Fig. 9a, designed to show the absolute pressure directly without adjustment. Essentially it is a barometer with only a very short tube provided for the mercury column. Instead, however, of having the mercury cup or reservoir at the bottom open to atmospheric pressure as in the ordinary barometer, the cup *C*, in this case, is sealed except for the pipe *P*, which is to be connected to the chamber in which the absolute pressure is to be measured. Before filling the instrument with mercury, the tube is exhausted to a practically perfect vacuum as for a regular-size barometer column. When the pipe *P* is open to atmospheric pressure, the mercury column will be forced all the way to the top of the capillary tube; but when this pipe is connected to a vacuum chamber, the mercury column will gradually fall in proportion to the vacuum in the chamber. If the instrument is well made, it should be fairly accurate for the range of absolute pressure between 4 pounds per square inch and $\frac{1}{2}$ pound per square inch. Graduations on the left side are inches of mercury and on the right pounds per square inch, both measuring absolute pressure.

Conversion of Pressures.—It is frequently necessary to reduce pressures in inches of mercury or of water to the equivalent in pounds per square inch. Since the weight of a cubic inch of mercury at 70°F. is 0.4906 pound and of water at the same temperature is 0.0360 pound, pressures in inches of mercury at the usual “room” temperatures can be reduced to pounds per square inch by multiplying by 0.491 or by dividing by 2.035, and similarly inches of water can be converted to pounds per square inch by multiplying by 0.0360 or by dividing by 27.78. Centimeters of mercury are reduced to pounds per square inch by multiplying by 0.1903.

Kilograms per square centimeter are reduced to pounds per square inch by multiplying the kilograms per square centimeter by 14.223 or by dividing by 0.0703. Grams divided by 28.35 are ounces avoirdupois; or **1 gram is approximately $\frac{1}{36}$ ounce.**

General metric conversion tables are given in Table V in the Appendix.

A cubic foot of water at 70°F. weighs 62.3 pounds and at 30°F., 62.4 pounds. At ordinary “room” temperature the pressure due to 2.31 feet of water is equivalent to 1 pound per square inch.¹

Tubes used as mercury manometers must be cleaned from time to time by washing the inside surface with nitric acid and afterward thoroughly cleansing them with water. Mercury used in manometers should be free from impurities. Usual impurities can generally be removed by filtering

¹ The unit pressure of 1 pound per square inch is equivalent also to that due to a column of air of uniform density, of which the vertical height in feet is approximately 144.0 divided by the weight of a cubic foot of air at the temperature, pressure, and humidity as observed. Tables of the weight of air are given on p. 187.

through a clean cloth of close texture or a very thin chamois leather. Air can be removed by boiling, but by far the best method of cleaning

mercury is by means of a mercury still. Unfortunately, an apparatus of this kind is not available in most engineering laboratories.

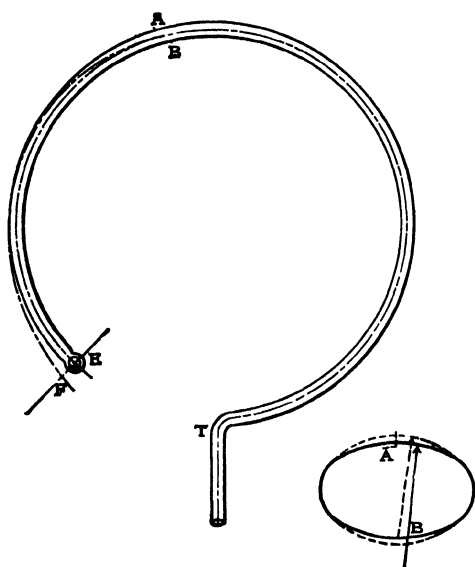


FIG. 10.—A typical Bourdon tube.

Pressure Gages.—The large size necessary for manometers or U-tubes, even if filled with the heaviest liquids, makes their use generally unsuitable except for comparatively low pressures. Instruments more desirable for high pressures are made by the application of some kind of elastic material designed to produce a uniform deformation for variations of pressure. By connecting a suitable auxiliary mechanism to the elastic element it can be

made to move a needle to indicate on a graduated dial the degree of pressure. The most common form of such devices is a hollow brass or steel tube bent into the shape of an arc of a circle.

It is a well-known principle that, when a straight piece of tubing is bent into this shape, the sides come nearer together, making the section of the tube a very much flattened oval.¹ A tube of this kind is illustrated in Fig. 10, which shows also in the right-hand corner a transverse section. If one end of such a tube is closed and fluid pressure is applied to the inside, the parallel sides, as at A and B, tend to separate, and consequently there is a tendency for the radius of curvature

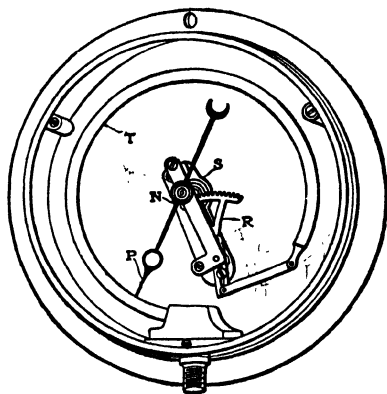


FIG. 11.—Mechanism of Bourdon tube gage.

of the tube to become larger, thus moving the end at E toward F. By connecting a suitable mechanism to E, the degree of pressure can be indicated. Instruments of this kind are called **Bourdon gages**.

¹ For a theoretical discussion of this principle in detail see *Theorie der Röhrenfeder Manometern*, Z. Ver. deut. Ing., Oct. 29, 1910, pp. 1865-1873.

Figure 11 shows one of the simplest forms of such gages used in power plants to indicate the pressures. It consists essentially of the curved tube *T* of oval cross section closed at one end. This end is connected by means of suitable levers to a rack *R*, engaging with a small pinion *N* on the same spindle with the pointer or needle *P*. Pressure applied to the tube *T* causes the rack and pinion to move over a dial

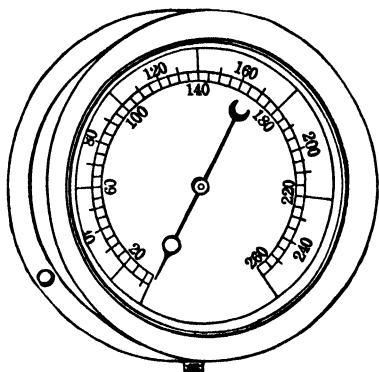


FIG. 12.—The dial of a pressure gage.

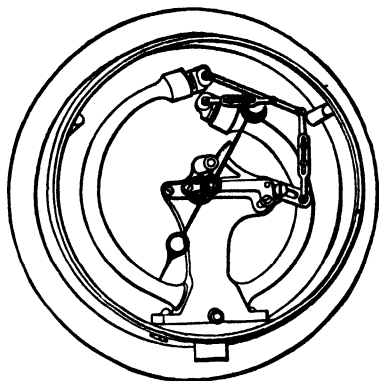


FIG. 13.—A modified Bourdon gage.

(Fig. 12) graduated or marked to indicate pressures in standard units, as, for example, pounds per square inch (English system) or kilograms per square centimeter (metric system). Gages are usually most sensitive and accurate from about one-half to two-thirds of the maximum graduation. The gage shown in Fig. 12 is most suitable for use between 130 and 175 pounds per square inch.

Figure 13 shows a form of Bourdon gage in which the amount of vibration of the needle due to the jarring that occurs in locomotive and other portable services has been reduced to a minimum by supporting the pressure tube in the middle instead of at its end as in Fig. 11. This form of tube has

also advantages for use in gages exposed to temperatures below freezing, since the arms can be drained of water, while the other form will usually hold the water that has entered.

In Bourdon gages any lost motion of the parts is taken up by the hair spring attached to the spindle carrying the pointer. But the use of a

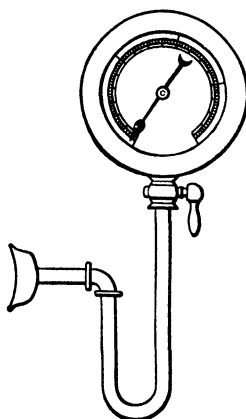


FIG. 14.

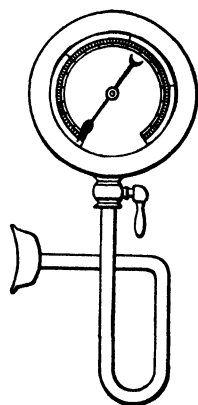


FIG. 15.

Examples of "goose-neck" steam siphons.

Bourdon pressure gage under conditions of constant vibration due either to moving over roads and fields on a tractor or to the fluctuation of pressure in a pipe due to the sharp cut-off of an engine will rapidly wear off the teeth of the rack *R* and of the pinion *N* (Fig. 11), so that it is not

unusual to find some gages very inaccurate in the portion of the scale most used.

In gages to be used to determine the pressure of ammonia, the oval tube is made of steel instead of brass, because the latter material deteriorates rapidly in the presence of ammonia.

Bourdon gages may be used for indicating the pressures of liquids, steam, or gases without taking any special precautions if the temperature is never much

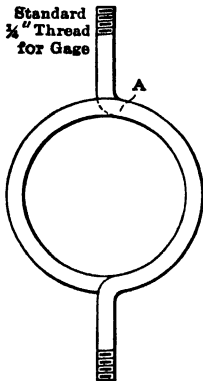


Fig. 16.

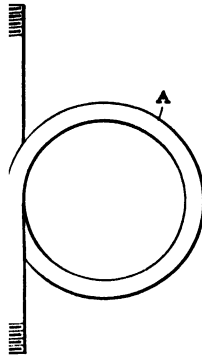


Fig. 17.

Examples of ring steam siphons.

over 150°F. If, however, the elastic tube in the gage is heated above this limit, it is likely to lose some of its temper. When used for steam pressure, therefore, **some form of siphon or water seal must always be used** to prevent steam from entering the gage. The forms of siphon shown in Figs. 14 and 15 are preferred for accurate measurements, but the types used most commonly are illustrated in Figs. 16 to 18. Ordinarily, brass pipe is preferred to iron, because of its better heat conductivity. This suggestion is particularly important for superheated steam, which requires a great deal of cooling in order to produce enough condensation to protect the gage from excessive heating. In the siphons shown in Figs. 16 and 17 there is always a possibility, however, that air carried in the steam may be entrapped at *A*, where it forms a "cushion" and reduces the water capacity of the siphon. Figure 18 represents a common and compact device for putting a water seal between a steam pipe and a gage.

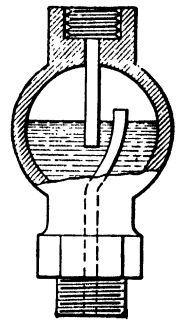


Fig. 18.—Simple water-seal steam siphon.

Adjustments.—The ratio of motion of the pointer with respect to that of the tube can be adjusted in most Bourdon gages by sliding a set screw in a slot in the short arm of the rack lever. In the gage illustrated in Fig. 11 when the short arm of the rack lever is made longer by adjusting the set screw, the movement of the rack and also of the pointer is reduced for a given deflection of the tube.

Sometimes when used carelessly, especially when subjected to pressures beyond the scale on the dial, the tube of the gage takes a permanent

“set”; or, in other words, it does not spring back to its original position, and the pointer does not come back to the zero mark. In such exigencies, and also for adjustment after calibration, the needle can be forced from its spindle—preferably by the use of a clamp or “needle-jack” made by gage manufacturers specially for this service—and then set again in position where it should be.

Another kind of gage in which there is a metallic disk or diaphragm instead of a bent tube for actuating the indicating device is sometimes used. A simple device of this type is shown in Fig. 19. It consists of a **corrugated diaphragm** *D* clamped around its edge by the flanges of an encircling chamber. Pressure applied on the lower side of the diaphragm deflects it upward, the amount of this

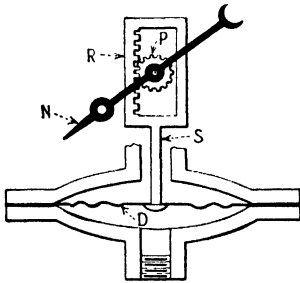


FIG. 19.—Principle of diaphragm gage.

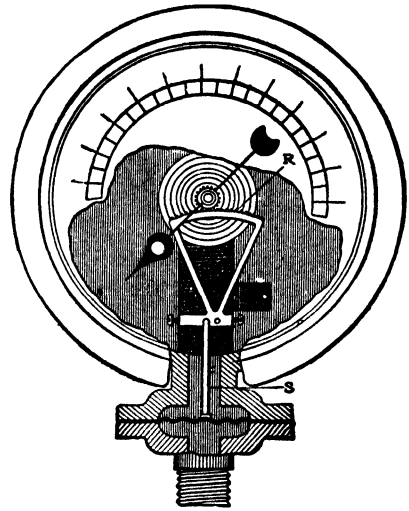


FIG. 20.—Typical diaphragm gage for measuring vacuum.

upward movement being proportional to the pressure. By means of a connecting strut *S* the movement of the diaphragm is communicated to a rack *R* connected to a small pinion *P* attached to the spindle of the needle *N* intended to indicate the pressure on a graduated dial. The gage shown in Fig. 19 is for measuring pressure. The one shown in Fig. 20 operates by the same principle but is used for measuring vacuum.

Since the deflection of the center of the diaphragm is proportional to the pressure and is inversely proportional to the cube of its thickness, a very slight alteration in the thickness of the diaphragm will cause a considerable change in the reading of the gage.

Hydrostatic pressure must be taken into account in some places, particularly when a Bourdon or diaphragm gage is used for determining the pressure in a boiler or in a water pipe. For example, if the gage is located below the surface of the water in a boiler, the gage will read too high, the error being the pressure equivalent to the head of water between the surface in the boiler and the center of a gage in a power plant. The correction in pounds per square inch is 0.432 times the head in feet. To make this correction unnecessary, the gage, if of the Bourdon type, should

be located so that its center will be at about the average water level in the boiler; and if it is of the diaphragm type, the diaphragm should be at about the average water level.

The practice of **choking the gage cock** to reduce vibrations of the pointer, unless done very carefully, is objectionable, especially when there is *leakage* around the joint of the plug in its cock. Under these circumstances the true pressure will be greater than that indicated.

Tapping a gage, preferably on the back, just enough to move the pointer noticeably, is an essential precaution before reading a gage having levers, racks, and springs in its mechanism, to be certain that all working parts are moving freely (see Report of Power Test Committee in *Journal of A.S.M.E.*, November, 1912, pp. 1695-1696).

Vacuum Gages.—For the measurement of vacuum instead of pressure, Bourdon gages are very commonly used. The design for a pressure gage is altered only in the arrangement of the levers moving the needle, which for vacuum measurements turn the needle in the same direction as for pressure (clockwise); but in this case the tube is bent inward or toward the center of the gage instead of outward as for pressure measurements. Vacuum gages are usually graduated to read **inches of mercury below atmospheric pressure**. **Absolute pressure in inches of mercury** is the difference between the barometer and the reading of such vacuum gages.

A vacuum gage is generally used to determine the **tightness of vacuum lines, condensers, etc.** If the line is tight, the gage will show no appreciable diminution of vacuum for several hours, but it drops quickly if there are leaks. These are most readily located by bringing a candle flame close to all the possible places of leakage. The flame will be drawn strongly toward the leaks by the current of air which is being drawn into the pipe by the vacuum inside.

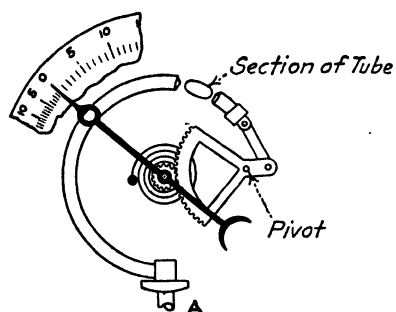


FIG. 21.—Compound gage.

Another type known as a **compound gage** is used to indicate both **pressure** and **vacuum** on the same dial. The linkages are adjusted differently from those in the usual types, in that the position of the pointer for zero gage pressure would now be at about the point marked 140 in Fig. 12. A compound gage is shown in Fig. 21. Gradua-

tions to the left around the dial would be from 0 to 30 inches of mercury (vacuum) and, similarly, to the right, from 0 to about 15 pounds per square inch. Such a device permits using a gage with a single Bourdon tube for measuring either vacuum or a considerable range of pressure.

Differential gages are designed to read pressure difference directly. Most commonly they are U-tube manometers (page 4) or equivalent devices arranged to have each side (or leg) connected to a source of pressure. The displacement of the liquid columns of the manometer will indicate the difference in pressure. In Fig. 190 (page 185) the U-tubes marked *a* are good examples of differential manometers or gages. Such

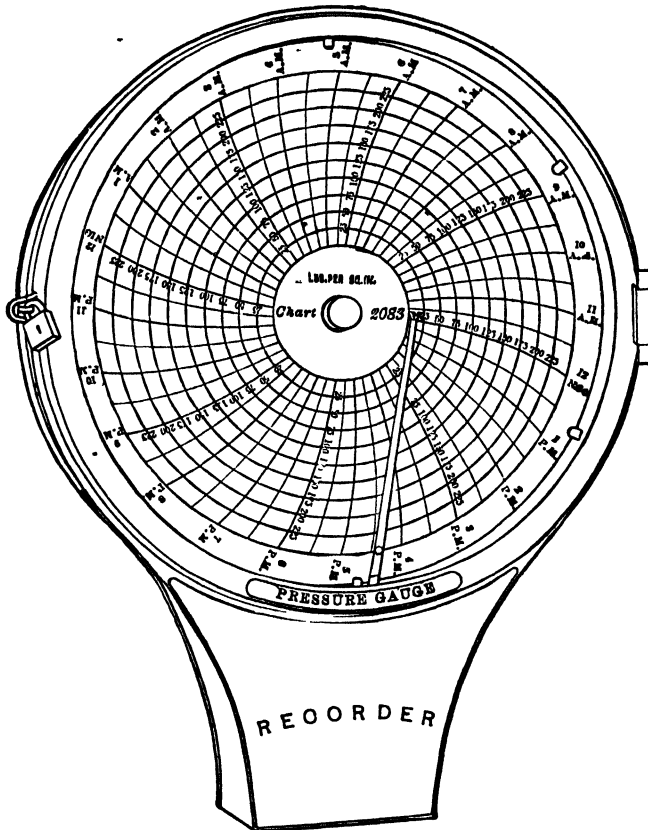


FIG. 22.—A recording pressure gage.

gages are very satisfactory for such purposes as measuring the difference in pressure on the two sides of an orifice in steam, air, or water pipes. Draft gages described on pages 28 to 32 are good examples of the differential type.

Recording Gages.—In many modern power plants recording gages are used to give a graphic record on a chart of the pressure or vacuum for 24 hours. The most common type of recording gage is shown in Fig. 22.

These gages are made usually in one of the three following forms: (1) with a tube of oval section (page 14) in the form of a helix, similar to the one shown in the recording thermometer illustrated in Fig. 53 (page 62);

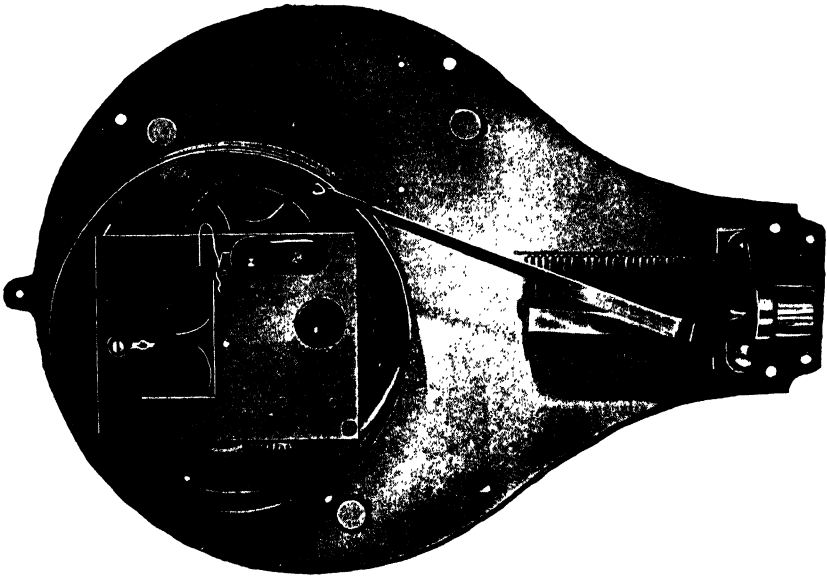


Fig. 24.—A low-pressure recording gage operated by "diaphragm" device.

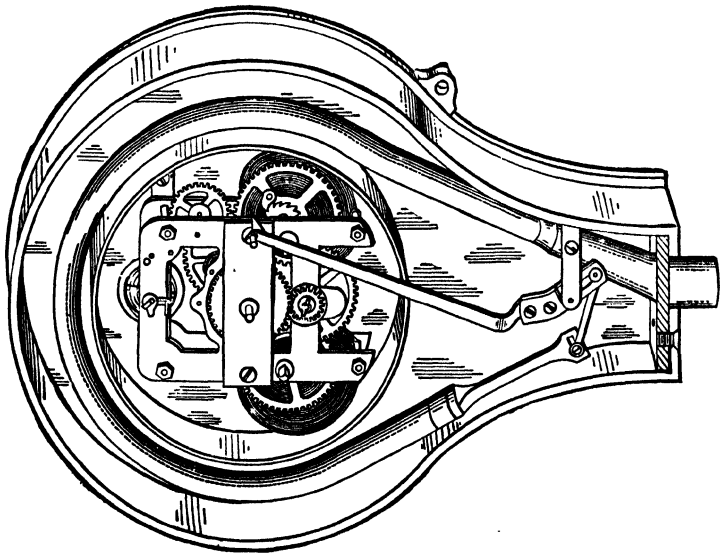


Fig. 23.—A recording pressure gage operated by a Bourdon tube.

(2) with an oval-section metallic Bourdon tube, as shown in Fig. 23; or (3) with a diaphragm or *syphon-bellows* device, as in Fig. 24. In the instrument shown in Fig. 24 the diaphragms spread out like an accordion when subjected to pressure on the inside. The first and second of these three types are generally used where the maximum pressure is greater than 3 pounds per square inch and the third when it is less. The recording arm is preferably attached directly to the moving element so that no gears, levers, or other multiplying devices are needed.

The **average pressure** corresponding to an irregular curve traced on the circular card of one of these recording gages is obtained with a fair degree of accuracy by integrating the curve by means of integrating instruments described on page 112. Corrections to be applied to the readings of these gages are, of course, obtained by calibrating in the same way as for an indicating gage.

Calibration of Gages.—Until recent years when the so-called “dead-weight” apparatus for testing gages came into general use, gages used in other places than engineering laboratories were commonly calibrated by **comparison** with a so-called **test gage**. Such test gages have usually somewhat finer graduations than the ordinary gages used in practice and are probably also adjusted a little more accurately. They should never be exposed to the severe conditions of service, being intended only for purposes of comparison. This comparison of pressure gages¹ can be made anywhere by connecting the standard and the gage to be tested to any system of piping in which the pressure can be varied either by pumping a liquid or by means of valves “throttling” steam, water, or air under pressure. The only important precaution to observe is that the two gages shall be at approximately the same level when a liquid is used, and that the velocity of the fluid in the main pipe to which the gages are attached is negligible or is the same at the points where the connections for the gages are inserted in the “main” pipe. Test gages must, of course, be calibrated from time to time with some standard apparatus to insure

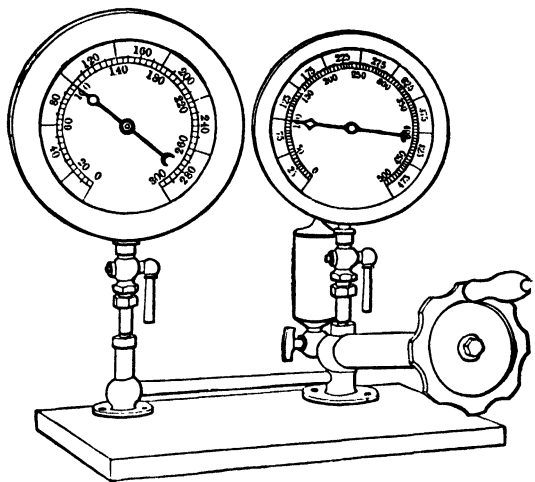


FIG. 25.—A bench test pump.

¹ Calibration of vacuum and low-pressure gages is discussed in another section (see p. 27).

their accuracy. A bench pump suitable for calibrating by comparison is shown in Fig. 25.

Gage Testers.—In very many power plants the test gage has been superseded by some form of gage tester and by this means the gages used in the plant can be calibrated directly with an absolute standard. Calibrations of gages for high pressures by means of mercury columns (page 24) are for practical reasons suitable only for laboratory work.

Dead-weight Gage Testers.—An apparatus for gage testing with dead weights is shown in Figs. 26 and 27. The latter figure shows a partial section. It consists of a vertical cylinder *C*, into which is fitted very accurately a plunger *P*, of which the area,

when new, is exactly one-fifth of a square inch. A circular platform upon which weights can be placed is attached to the upper end of this plunger. The cylinder *C* communicates at its lower end with the reservoir *R* fitted with an adjustable piston working in a screw and is operated by a hand wheel. A pipe *T* attached to the lower part of the reservoir is provided with unions and special fittings for attaching gages of various ranges. In the horizontal portion of this pipe there is a three-way cock (page 149) or valve *V* either for draining the reservoir or for closing the pipe so that the liquid in the apparatus will not escape when the gage is removed. In operation, after the gage has been attached securely, the piston is screwed down to the bottom of the reservoir *R*, then, with the plunger *P* removed, glycerine or heavy oil is poured into the cylinder *C* at the same time that the piston is screwed out. In this way the reservoir can be completely filled with oil without entrapping any considerable amount of air, which would act as a cushion preventing the most satisfactory operation of the apparatus.

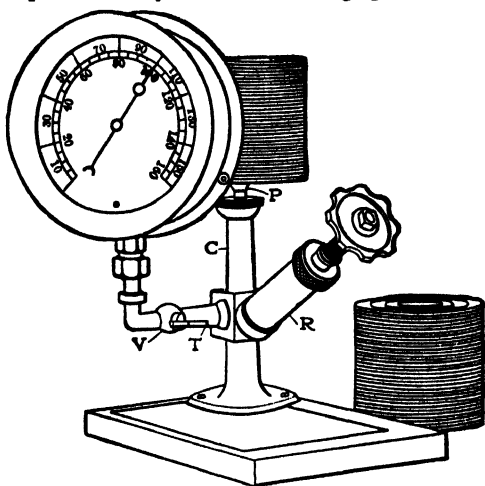


FIG. 26.—Crosby dead-weight gage tester.

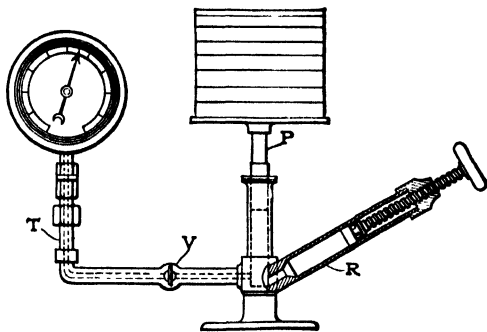


FIG. 27.—Section of a dead-weight gage tester.

If the area of the plunger P is one-fifth of a square inch, then each pound weight added on the platform produces a pressure on the liquid of 5 pounds per square inch. The weight of the platform and plunger (usually 1 pound) must always be included in the weight producing the pressure. As the load on the platform is increased, the piston must, from time to time, be screwed in to keep the platform "floating." When observations are being taken, it is essential that the loaded platform be given—preferably by hand—a slight rotary motion to reduce to a minimum the friction of the plunger in its cylinder.

Suggested Procedure with Dead-weight Testers.—The accuracy of the gage to be calibrated is determined by subjecting it to known pressures and noting its error. Before the plunger P has been put into place, the reading of the gage, called "zero reading," should be observed and recorded in a form similar to the one on page 26. Then the pressure should be increased 5 pounds per square inch at a time (corresponding usually to a weight of 1 pound) up to the limit of the graduations on the dial, **spinning the piston gently when each reading is taken.** Commencing then with the highest pressure, the same operation should be repeated by decreasing the pressure by the same increments.¹

In case there is an appreciable difference between the area of the plunger P and its cylinder C , the two areas should be averaged to obtain the true area to be used in calculating the unit pressure. In other words, the true pressure exerted on the fluid in the apparatus in pounds per square inch is the **total weight in pounds divided by this average area in square inches** (Report of Power Test Committee of A.S.M.E., November, 1912).

A modification of the dead-weight gage tester is shown in Fig. 28. This instrument is particularly suited for calibrations at high pressures. Its range is from 0 to 1,500 pounds per square inch. Any pressure within these limits can be obtained without shifting heavy weights. Readings are taken when the scale beam is balanced. The hand wheel A

¹ When the pressure is being decreased, the movement of the pointer must be always in a counterclockwise direction just before a reading is taken. In other words, if a weight of 2 pounds has been taken from the load on the plunger when only 1 pound should have been removed, the pointer will, of course, get below the next point to be calibrated. To secure the reading missed, it will not be correct to add 1 pound and take the reading, because the friction and lost motion will now be in the same direction as with increasing pressures; and to overcome this difficulty the pressure must be increased again to a value higher than that for which the reading is to be taken. For the purpose of increasing the weight it is not necessary to put on more weights, as additional load in such cases can be put on by the pressure of the hand.

The same precautions apply with even greater force to calibrations made with test gages or with a mercury column. With either of these instruments, discrepancies may occur with increasing or decreasing pressures. In fact, the only certain way to get satisfactory results with these instruments is to keep the pointer of the gage or the mercury column, as the case may be, moving continually in the same direction.

is used to regulate the fluid pressure by means of a piston as in the apparatus shown in Figs. 26 and 27. The other hand wheel *B* shown in the figure must be kept rotating when observations are taken. The slight jarring of the parts due to its rotation serves to make the friction as small as possible. Particularly in gages badly worn from severe usage, it is necessary to tap the back of the casing of the gage lightly before each reading is taken to be certain that the pointer is moving freely.

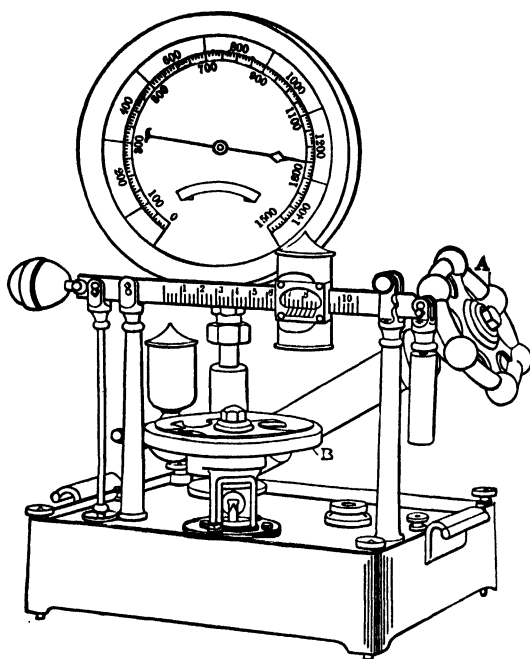


FIG. 28.—Crosby portable fluid pressure scales.

Calibration of Gages with Mercury Columns. The ultimate standard for the determination of reasonably high pressures is the mercury column, but the apparatus required is so complicated and occupies so much space that this method is suitable only for use in laboratories where it will have the attention of skilled observers.

For calibrating steam gages, mercury columns have been fitted up in a variety of ways. The simplest method connects the gage to be tested, by means of a short tube, to a "closed" mercury well, into the top of which a long glass tube has been inserted. The pressure can then be increased by displacing some of the mercury in the well by means of the plunger in the mercury pump, shown at the right-hand side of Fig. 30, and forcing it up into the glass

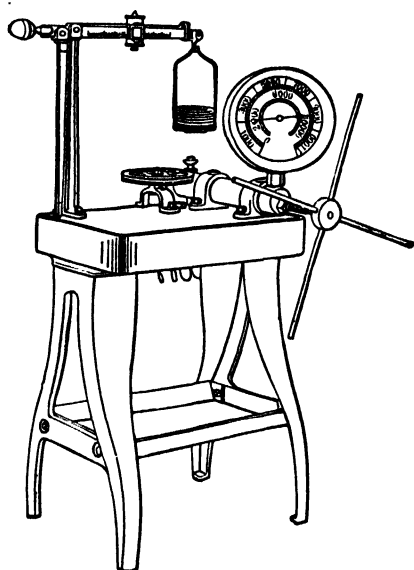


FIG. 29.—Fluid pressure scales for high pressures.

tube. Zero pressure for comparison is to be taken on the column at the same level as the center of the gage. Beginning then with 5 pounds per square inch pressure on the gage, observe the corresponding height of the mercury column and its temperature, and then continue the observations, first continually increasing the pressure and then decreasing by increments of 5 pounds, as indicated by the gage. Equivalent units for calibration can be computed from the height of the mercury column, since 1 inch of mercury at 70°F. is equivalent to a pressure of 0.4906¹ pound per square inch.

When a mercury-testing apparatus is used, it is necessary to observe the temperature near the mercury column in the room in which the work is being done, so that the observed height of the mercury column can be corrected to a temperature

at which the relation between pressure in pounds per square inch and height is known. The coefficient of cubical expansion of mercury is not constant, as will be observed in Table IV.

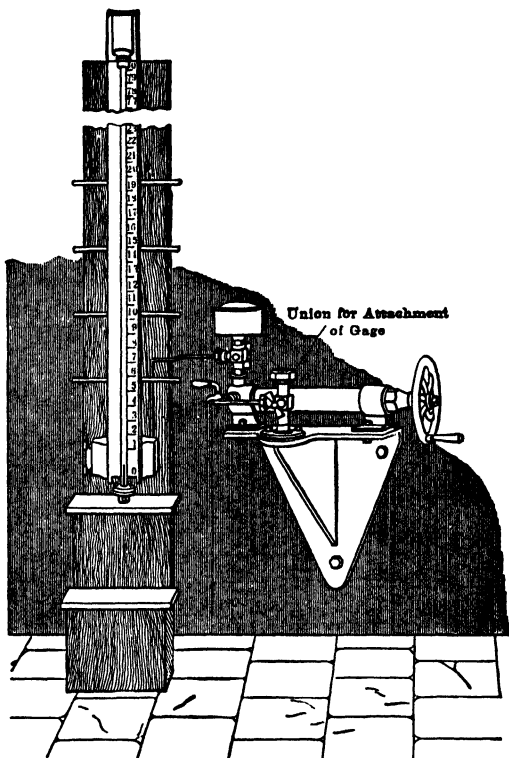


FIG. 30.—Standard mercury column and hand pump.

TABLE IV.—COEFFICIENTS OF CUBICAL EXPANSION OF MERCURY

Temperature, Degrees Fahrenheit	Coefficient of Cubical Expansion
32	0.0000998
50	0.0001000
70	0.0001002
90	0.0001004
110	0.0001007

For very accurate work, allowance must be made for the linear expansion of the graduated scale. Coefficients of expansion of metals are given in the Appendix, Table IV.

¹ This value must not be confused with the weight of a cubic inch of mercury at 32°F. for barometer correction (p. 8), which is 0.4912 pound.

Observations taken in the calibration of a steam gage should be recorded and the computed errors tabulated in a form similar to the following:

CALIBRATION OF PRESSURE GAGE COMPARISON WITH GAGE TESTER

Date.....Observers.....
 Maker of gage.....Maker's No.....
 Laboratory No.....Limits of graduation.....

No. of reading	Weight on tester, pounds	Gage readings, pounds per square inch			Actual pressure, pounds per square inch	Mean error of gage, pounds per square inch ¹	Remarks
		Up	Down	Mean			

¹ When the sign is +, the correction ("error") should be added, and when -, should be subtracted from the observed reading.

The error of the gage is determined by the comparison of the mean of the up and down readings with the actual pressure.

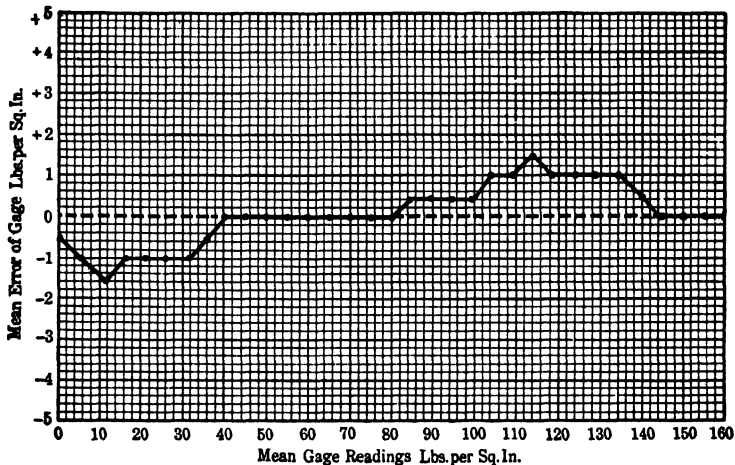


FIG. 31.—Typical error curve for a pressure gage.

Curves.—From the data tabulated above two curves are usually plotted:

1. **Mean gage readings** (abscissas) and actual pressures (ordinates). Use a large sheet of coordinate paper for this curve. Unless plotted to a very large scale, however, such a curve will be of little value.

2. **Error Curve.**—Mean gage readings (abscissas) and mean corrections, positive and negative (ordinates) (see curve in Fig. 31). **Error curves** should have the points, if very irregular, connected by a broken line rather than by a "fair" or average curve through them. Never, however, try to draw an irregular curve through each of a number of scattered points when the points are supposed to follow a definite relation or law between the coordinates selected. A "fair" curve should then be drawn **between** the irregular points.

Hysteresis Calibration Loop.—When both increasing and decreasing readings of an instrument are plotted on a curve sheet against true or standard values, a so-called "hysteresis loop" is obtained if the work of calibration is done carefully. An example of such a loop is shown in Fig. 32. The increasing ("ascending") values were obtained first and, when plotted, made the curve *abc*. Similarly, decreasing ("descending") values made the curve *cde*.¹ In such a case the lower end of the curve will not usually close. The difference between the increasing and the decreasing values is caused by friction and lost motion. The calibration curve is found by plotting a series of mean values, such as *f* in the figure which is the mean of the readings at *b* and *d*.

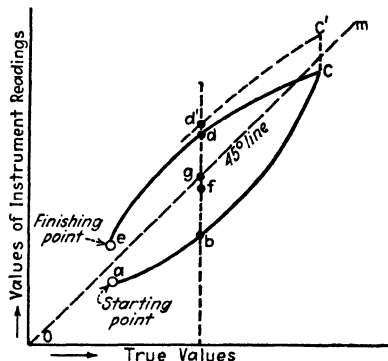


FIG. 32.—Hysteresis loop of instrument readings.

An ideally perfect instrument will have no hysteresis loop in its calibration curve, and both increasing and decreasing readings would fall on a 45-degree line *om* in Fig. 32.

If increasing values had been carried *beyond* the point *c* and then decreasing values had been observed as at *c'*, the curve *c'd'e* would have been obtained, the value of *c' - c* being the friction and lost motion in the instrument at the highest *observed* value.

Calibration of Vacuum and Low-pressure Gages.—A vacuum gage is usually calibrated by connecting it to one end of a U-shaped glass tube both legs of which are about 30 inches long and are filled to about half their length with mercury. The U-tube and the gage are then connected

¹ Obviously, if calibration values are to be used for correcting indicated values of the instrument that are near *c* on the curve, a more accurate method is to use increasing values larger than at *c*, and from that higher value begin the curve of decreasing values.

to the receiver of an air pump or to an aspirator or ejector operated by water or steam pressure, such as chemists use for vacuum filtering. The aspirator (Figs. 240 and 241, page 267) is really the more convenient instrument to use. If the readings of the vacuum gage are correct, they will correspond exactly with the difference in the level of the mercury in the two legs of the U-tube.

In case a condensing engine is operating when the calibration of the vacuum gage is to be made, both the gage and the glass U-tube may be connected to the condenser. A comparison of the readings taken will show, under the best possible conditions, the absolute errors of the gage. A suitable scale about 30 inches long and accurately graduated should, of course, be provided and placed between the two legs of the U-tube.¹

A low-pressure gage with a scale from, say, 0 to 15 pounds per square inch is very easily and accurately calibrated by using the same glass

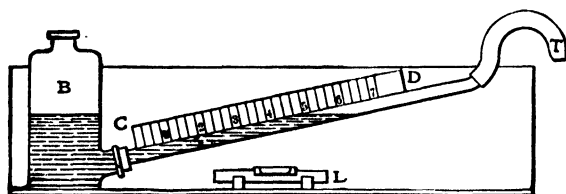


FIG. 33.—Inclined tube draft gage.

U-tube mentioned for the calibration of the vacuum gage with air **pressure**, preferably, or with steam pressure. Otherwise the method of calibration is the same as for a vacuum gage, except that inches of pressure instead of inches of vacuum are observed.

Draft Gages.—Many engineers use an ordinary glass U-tube manometer filled with water for measuring small pressures like that due to the draft in a chimney or that produced in air ducts by ventilating fans or blowers. For such observations in many cases, however, greater accuracy is desired than can be secured with the ordinary U-tube like Fig. 2, and a special form of manometer is used in which the distance moved by the surface of the liquid in the tube is greater than the vertical change of level. Figure 33 illustrates a simple device of this kind. It consists of a bottle *B*, filled with water, having a suitable opening at the bottom to which the inclined glass tube *CD* is attached by means of a short rubber tube. At the upper end of this tube a piece of rubber tubing *T* is shown and is intended to be connected to the chimney, duct, or flue in which

¹ When measuring inches of vacuum by means of a single mercury column dipping into a cup, the zero is to be taken at the level of the mercury in the well into which the glass vacuum tube enters. The elevation of the gage above the surface of the mercury in the well is not to be considered, as the weight of the column of air between the center of the gage and the level of the mercury is negligible. For the use of vacuum gages on water suction pipes, see sections on Hydraulic Machinery (Chap. XXIII).

the pressure is to be obtained. A scale placed behind the inclined tube CD should be graduated so that, when the spirit level L is adjusted, vertical differences in level of the liquid in CD will be indicated by its graduations. Then differences in the readings of the scale will give directly the difference in pressure in inches of water just as with an ordinary U-tube. Unless the bottle B is very large in diameter, it is best to mark the graduations on the scale for every half inch by comparison with a U-tube¹ as in Fig. 2. Intermediate divisions can then be marked with the help of dividers.

Theory of Inclined Draft Gage.—Draft gages are made with one leg inclined in order to increase the accuracy of the readings. For example, the readings taken from the draft gage in Fig. 2 can be read accurately to the nearest tenth inch and may be estimated to hundredths of an inch; while if the inclined manometer shown in Fig. 33 is properly adjusted and calibrated, the readings are accurate to the nearest hundredth of an inch and may be estimated to thousandths of an inch. Briefly, the reason for this greater accuracy of readings from the inclined type in comparison with the ordinary manometer is that for the same vertical height a slanting line connecting two points at different elevations is longer than a vertical line that could measure an equivalent difference in elevation. More accurately, the theory of the inclined manometer with a reservoir R may be expressed by the following notation, thus:

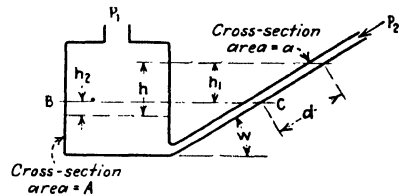


FIG. 34.—Diagram to explain theory of inclined manometer.

thus:

A = cross section of reservoir R , in square inches.

a = cross section of inclined capillary tube, in square inches.

P_1 = absolute pressure in reservoir R , in pound per square inch.

P_2 = absolute pressure in inclined tube, in pound per square inch.

h_1 = vertical distance the level in reservoir R rises when the pressure P_2 is slightly greater than P_1 , in inches.

h_2 = vertical distance the level in reservoir R falls when the pressure P_2 is slightly greater than P_1 , in inches.

h = total vertical displacement of level = $h_1 + h_2$, in inches.

d = rise of liquid in inclined leg, measured along tube, in inches.

Obviously, a pressure difference $P_1 - P_2$ produces a change of level in the inclined leg (angle w in Fig. 34) or $P_1 - P_2 = h = h_1 + h_2$.

Since the volume of liquid entering or leaving the reservoir R must be the same as the volume leaving or entering the inclined leg,

$$h = d \left(\sin w + \frac{a}{A} \right)$$

¹ This method for graduating and also for checking the graduations is recommended by the Power Test Committee of the A.S.M.E. For this special service the use of alcohol is advised instead of water in the U-tube, but allowance must be made for the difference in specific gravity between water and alcohol. It would be impracticable to use alcohol in gages that are in continuous service, as the ordinary draft gage must be open to the air at one end, and the alcohol would very rapidly vaporize.

The *multiplying factor* of the inclined manometer is, then,

$$\frac{d}{h} = \frac{1}{\sin w + \frac{a}{A}}$$

This equation shows that by reducing the angle w the multiplying factor will be proportionally increased. The smaller the angle w is, the greater the *sensitivity* of the manometer will be, but *accuracy* may not be increased in the same proportion because the inaccuracies that are caused by sluggishness due to capillarity and difficulties in sufficiently exact leveling will be increased. Since alcohol has much less sluggish capillarity than water, these troubles are not significant if alcohol is used as the liquid in the manometer and the angle w is not much less than 10 degrees.

Only small pressure ranges are possible with the ordinary type of inclined manometer. For large ranges several manometers of varying ranges may be used in combination, or a special manometer with an adjustable leg, of the kind used in aeronautic work,¹ may be suitably employed.

Very accurate draft gages of this type, known as Ellison's, are shown in Figs. 34a and 34b. The inclination of the tube in these instruments is

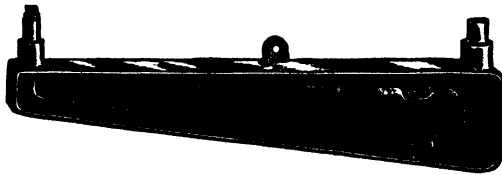


FIG. 34a.—Ellison's improved draft gage.

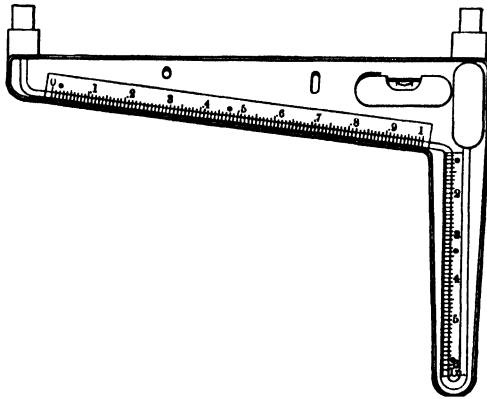


FIG. 34b.—Ellison's differential direct-draft gage for high drafts.

usually about 1 to 10. Instead of water or alcohol, a very light oil is used to fill the tube. It is claimed that this oil has the advantages of having less capillarity than water and has less evaporation than either water or alcohol. Graduations on these instruments which are sold commercially² are, however, always made to read equivalent inches

¹ Manometers with an adjustable leg are explained in Diederichs and Andrae, "Experimental Mechanical Engineering," Vol. I, p. 78.

² American Steam Gage and Valve Manufacturing Co., Boston and Chicago.

of water. These draft gages are also often used for measuring small differences of pressure. For example, if there are two vessels containing gases at different pressures and one is connected to the left-hand side and the other to the right-hand side of the gage, it will indicate the difference in pressure. When used in this way, it is called a *differential gage* (see page 19).

Graduations on all such instruments should be checked by comparison with a simple U-tube, filled with **distilled water** (condensed steam from a surface condenser is satisfactory). The U-tube should be provided preferably with a steel scale of a recognized standard make. A simple device for calibrating is shown in Fig. 35. The draft gage to be tested and the U-tube (standard) are connected by rubber tubing to a T-piece *B* of hard-rubber or glass tubing to which suction in varying amounts can be applied by putting the free end of the tube in the mouth with the

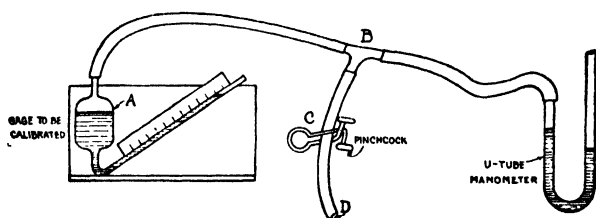


FIG. 35.—Calibrating device for draft gages.

pinchcock open and then closing the pinchcock when the desired suction is obtained.

When gages are calibrated, it is worth while to notice that, when instruments are to be used to observe practically constant values, it is necessary to calibrate them only near the values to be observed.

Draft gages which have liquids of slightly different densities combined so as to magnify the difference in level are sometimes used. Generally such an instrument consists of a U-tube with the top ends of very large area¹ and the connecting portion of small capillary tubing. Obviously, the liquids used must have very little tendency to mix together or to become diffused. Figure 36a shows one type of such gages. As illustrated, the U-tube is first partly filled with the heavier liquid, and afterward a lighter liquid is poured into each leg so as to fill completely the capillary and also fill **partly** each of the vessels *A* and *B*. The level of the separating meniscus on either side can be used as the indicator. Since the area of the capillary is very small compared with that of the vessels *A* and *B*, the indicating meniscus receives quite a large displacement for small differences in pressure. If the liquids used are kerosene (specific gravity = 0.80) and dilute alcohol (specific gravity = 0.85), the displacement of the separating meniscus due to pressure or vacuum will be about

¹ Preferably these top ends should be made with constant sectional area.

twenty times the equivalent inches of water. The alcohol may be colored with an aniline dye that is insoluble in kerosene. Instruments of this type are usually calibrated and graduated to indicate the pressure in inches of water.

A very delicate draft gage can be made also by using the apparatus of Fig. 36a in a little different way, as in Fig. 36b. Water is put into *A* and oil into *B*, with the surface of separation originally at *Q*, when *A*

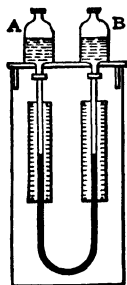


FIG. 36a.—Simplest two-fluid type of manometer.

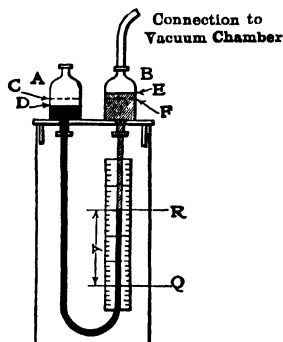


FIG. 36b.—Another type of two-fluid manometer.

and *B* are both subjected to the same pressure. When *B* is connected to a chamber in which there is a vacuum as in the breeching or the chimney of a power plant, the levels are changed from *F* to *E* in *B* and from *C* to *D* in *A*. The surface of separation moves from *Q* to *R*. Let $CD = EF = x$ inches, $QR = y$ inches, and s = specific gravity of the oil, then the change of pressure p in inches of water in *A* and *B* is

$$p = x + sx + y(1 - s).$$

If the areas of *A* and *B* are equal and are n times the area of the connecting tube, then $y = nx$, and $p = \frac{y + sy}{n} + y(1 - s) = y \left[\frac{1 + s}{n} + 1 - s \right]$.

Observe that the smaller the factor in the brackets the greater will be the magnification or sensitiveness. The value of n should therefore be made as large as possible, say about 100, and the two liquids should be preferably of nearly the same specific gravity. Gasoline and denatured alcohol diluted with water and colored are frequently used for the two liquids. They can be made of nearly the same specific gravity and the line of separation is readily observable. If S is the specific gravity of the alcohol and s that of the gasoline, obviously

$$p = y \left[\frac{S + s}{n} + S - s \right].$$

Values of pressure causing a given movement of the line of separation may be calculated from the equations above, but actual calibration by

comparison with an ordinary U-tube and graduation by extrapolation as recommended by the A.S.M.E. Test Codes are always preferable.

These last types of draft gages are too delicate for ordinary service in power plants to measure draft but are particularly well suited for use with Pitot tubes (page 184) in measuring air velocities.

CHAPTER II

MEASUREMENT OF TEMPERATURE

Mercurial Thermometers.—Temperatures less than about 500°F. are usually measured by means of mercurial thermometers, depending for their action on the expansion of mercury in a glass bulb and a graduated capillary tube.¹ The ordinary type is made so as to have merely a vacuum in the capillary tube above the mercury. For use at higher temperatures the capillary may be filled with nitrogen gas when the thermometer is made. It will then be serviceable up to 1000°F. Thermometers made of quartz instead of glass and filled with nitrogen or carbonic acid gas can be used for temperatures as high as 1500°F.² Quartz thermometers are much stronger than those made of glass but are too expensive for ordinary commercial use.

A.S.M.E. Tests of Temperature-measuring Instruments and Apparatus.³—Under many of the conditions met in power tests the desired accuracy in the measurement of temperature can be obtained only by observance of suitable precautions in the installation and use of temperature-measuring instruments, and in the interpretation of the results obtained with them.

Many of the instruments available for temperature measurements are capable of indicating temperature far more accurately than is required in most tests. The difficulties in the use of the instruments are due either to wrong installation or to careless use.

¹ The best thermometers for ordinary engineering work have graduations etched on the stem and filled with ink. This is the type of mercury-in-glass thermometer recommended by the Power Test Committee of the A.S.M.E. After considerable use, however, the ink originally in the etched markings disappears and it becomes difficult to read the scale. When this happens a thick paint made of lampblack and shellac or printer's ink can be rubbed over the etched scale, and when this paint is rubbed off, after a few minutes, there will be enough of it left in the etchings to make the scale as legible as when new. A crayon or pencil of soft greasy graphite, like those used by glaziers for marking glass or by shippers for marking cases, is a satisfactory substitute for the paint, although it is not so permanent.

² The boiling point is dependent upon the pressure on the liquid, and in thermometry the temperature range is often extended by introducing an inert gas such as nitrogen into the capillary space above the liquid, thus increasing the pressure on the liquid.

³ Abridged from A.S.M.E. Power Test Codes, Instruments and Apparatus, Part III, (Approved by Council, February, 1931 and articles in *Mech. Eng.*, March, April, and May, 1926.

One of the principal sources of error in temperature measurement is the natural tendency toward temperature equalization. This tendency which causes the flow of heat from regions of higher to regions of lower temperature must always be borne in mind, and the effect of any possible flow on the reading of the instrument must be determined. For example, if a thermometer is inserted in a pipe (Fig. 37a) in which superheated steam or a highly heated gas is flowing, it is commonly assumed that the thermometer acquires the temperature of the steam or gas. As a matter of fact, with temperatures between 500 to 600°F. inside the pipe and temperatures of 50 to 100°F. outside the pipe, there may be an error in temperature measurement varying from 1 or 2° to as much as 40 or 50°F., the thermometer indicating too low a temperature.

Another example of temperature errors is furnished by thermocouples. For example, if a thermocouple has a metallic protecting tube which projects into a vessel, duct, or pipe containing a fluid at a temperature lower than the surroundings, heat will flow along the leads of the thermocouple and along the protecting tube from the surroundings to the fluid and the thermocouple may have a temperature higher than that of the fluid being measured.¹

Another source of error in the measurement of temperature during a test is due to failure to take account of a variable rate of flow of the material the temperature of which is being measured. As an example, assume that the average temperature of the feed water entering a boiler

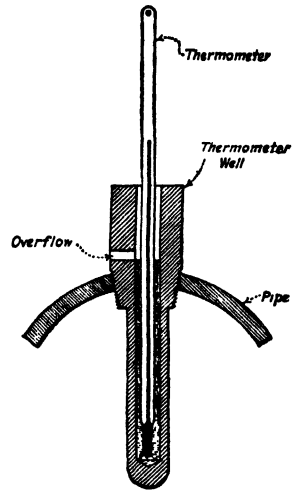


FIG. 37a.—Plain thermometer well.

¹ Neglect to consider mass flow when measuring the average temperature of fluids flowing in a conduit may result in an error of another nature. As an example, assume that it is desired to determine the average temperature of hot gases flowing in a horizontal direction through a flue of large cross section, say, 10 by 10 feet. Assume that preliminary tests have shown that there is quite a variation in temperature of gas between different elevations in the flue and that the engineer has decided to obtain the average temperature by stringing a wire back and forth across the section of the flue and using this wire as a *resistance thermometer* (p. 65). This would seem at first sight to meet all the requirements, but the possibility of error is very large. If it is assumed that the wire does correctly average the temperature over the cross section—an assumption open to serious question—a large error is still possible. In all probability the higher temperatures will be found at the top of the flue, and it may be that the connections are such that the velocity of gases at the top of the flue will be much greater than that of gases at the bottom. The average temperature of the gases flowing through the flue will under such conditions be higher than the average temperature of the cross section, if mass flow is not taken into account.

is to be measured during a boiler test, and that the feed pump is practically shut down while the fires are being cleaned. Obviously, the measurement of the temperature of the feed water at stated intervals of time will not give data for determining the average temperature of the water entering the boiler, if the water temperature is greatly different at low rates of flow from what it is under normal conditions. In such a case the temperature should be measured every time that a stated amount of water has passed a certain point in the feed system, as, for example, every time, or every second time, that the tanks for weighing feed water are emptied.

The reduction in the rate of flow of a fluid, especially in small pipes, may produce a considerable error in the observed temperature. In fact it is possible to have an *increase* in the apparent temperature of feed water as much as 100 or even 200°F. during a period of very small flow in the pipe. Sometimes it is impossible to obtain the correct temperature in a *boiler-feed pipe running near and parallel* to a boiler setting. The reason for this is that the conduction of heat from the boiler along the feed-pipe walls is likely to increase considerably the reading of a thermometer located in the pipe, without affecting to the same extent the temperature of the water flowing at low velocities through the pipe. In a case of this kind, a temperature measurement with a mercury-filled thermometer and well should be avoided and, if a permanent installation is desirable, a resistance thermometer or a *thermocouple* may be used with less error.

Similar difficulties in the measurement of the temperature of gases in ducts where the velocity is quite low may be remedied by artificially creating a relatively high velocity of flow of the gases past the temperature-measuring element. By thus increasing the velocity, the errors due to radiation may be reduced.

Types of Temperature-measuring Instruments.—Mercury- or alcohol-filled thermometers are ordinarily used for low-temperature measurements and for this reason in low-temperature work one ordinarily thinks of a thermometer as the suitable temperature-measuring instrument. Without apparently good reason, instruments for measuring high temperatures are commonly spoken of as *pyrometers*, the fact that *electric-resistance thermometers* may be suitable for very low as well as very high temperatures being neglected. The following types of temperature-measuring instruments are recommended by the Power Test Committee of the A.S.M.E. for use in testing: (1) *Thermocouple pyrometers* (page 63), in which the voltage generated at the junction of two dissimilar metal wires indicates the degree of temperature, the voltage at the junction increasing proportionally with the temperature. (2) *Resistance thermometers* (page 65), in which the resistance of a calibrated wire changes with the temperature, the resistance change being proportional to the increase in temperature. (3) *Liquid-filled glass thermometers*, in which there is an expansion or contraction of a liquid corresponding to changes in temperature, the expansion of the liquid being proportional to the increase in temperature. The liquids commonly used in such

thermometers are mercury, alcohol, or pentane. (4) *Bourdon tube thermometers* (page 59), which operate by the expansion of a fluid (liquid or gas) as follows: (a) expansion of a liquid that completely fills the enclosed tubing and bulb of the instrument, (b) expansion of the liquid in the *bulb* of the instrument, (c) expansion of a gas that completely fills the tubing and bulb of the instrument. (5) *Radiation pyrometers* (page 67), in which there is a small body capable of absorbing radiation of all wave lengths, the radiation absorbed being proportional to the temperature. (6) *Optical pyrometers* (page 69), by which temperature is determined by matching the luminosity of the hot body of which the temperature is to be determined with the luminosity of a calibrated source of light. (7) *Pyrometer cones* (page 70), by which temperature is determined by the bending over of a graded set of ceramic cones, each having a definite heat-resisting value. (8) *Bimetallic thermometers* (page 66), which depend for their operation on the differential expansion of two solids (usually metals), the differential expansion being proportional to the increase in temperature. The Power Test Code Committee of the A.S.M.E., however, does not recommend bimetallic thermometers.

Accuracy of Temperature-measuring Instruments.—The recommended types of temperature-measuring instruments described in the last paragraphs are listed as in Table V for temperature ranges and limits of error.

For low-temperature work mercury thermometers are generally preferred, as they can be made to almost any degree of accuracy required. When provided with a delicate galvanometer, electric-resistance thermometers can be used with a very high degree of accuracy, and, in fact, temperature differences can be determined with them very much more accurately than with the best mercury thermometers. Next in degree of accuracy to electric-resistance and mercury thermometers are probably thermoelectric pyrometers; and it is interesting that a pyrometer of this kind can be readily made by twisting together at their ends rods of wrought iron and nickel. It is essential that the ends should be welded, as welding (preferably electric) gives the couple greater permanency, by preventing the accumulation of dust from interfering with electrical conductivity. The loose ends can be connected up to the binding posts of a millivoltmeter by insulated copper wires and calibrated. The only disadvantage will be that it will not have a scale reading directly in degrees of temperature. The wrought-iron and nickel rods should be covered with a winding of asbestos tape to keep them separated.

Calibration of Temperature-measuring Instruments under Conditions of Service.—Whenever possible, the calibration of a temperature-measuring instrument should be made as nearly as possible under the conditions of actual service. For example, a mercury-filled thermometer should be calibrated in the same thermometer well in which it is to be used in tests

TABLE V.—TEMPERATURE RANGE AND ERROR LIMITS OF THERMOMETER TYPES

Type	Temperature range, degrees Fahrenheit	Limits of error, degrees Fahrenheit
Thermocouple pyrometers:		
(a) Base metal.....	300 to 2,000	2 to 20 (base metal and rare metal)
(b) Rare metal.....	300 to 2,800	Limits of error depend on indicating instrument
Electric-resistance thermometers.....	—300 to 1,800	0.005 to 5 Depend on indicating instrument
Liquid-in-glass thermometers:		
(a) Ordinary glass, mercury filled.....	—35 to 750	0.5 to 7
(b) Corning or Jena glass and nitrogen filled.....	—35 to 925	0.5 to 7
Bourdon-tube thermometers:		
(a) Liquid-filled type:		
Alcohol filled.....	—50 to 300	2 to 10
Mercury filled.....	—38 to 1,000	2 to 10
(b) Vapor-pressure type:		
Alcohol filled.....	200 to 400	2 to 10
Ether filled.....	100 to 300	2 to 10
Sulphur dioxide filled.....	20 to 250	2 to 10
Aniline filled.....	400 to 700	2 to 10
Methyl ether filled.....	—20 to 200	2 to 10
(c) Gas-filled type:		
Nitrogen filled.....	—60 to 1,000	2 to 10
Radiation pyrometers.....	Over 1,000	20 to 30 for black-body conditions
Optical pyrometers.....	1,500 and up	15 to 35
Pyrometer cones.....	1,100 to 3,600	About 15 to 30 in best makes

so that the exact conditions of emergent stem exposure (page 45) and temperature of surrounding air will be duplicated. Obviously, for calibration, approximately the same accuracy can be obtained if an *identical* thermometer well is used.

For temperature measurements of highly heated gases or superheated vapors, finned thermometer wells (page 39) are required in order to obtain accuracy in temperature measuring. The fins on such wells will compensate to some extent for the lower rate of heat transfer from a gas or superheated vapor to the bulb within the well as compared with the rate of heat transfer from a liquid or from a saturated vapor to the bulb within the well.

Radiation of heat between bodies at different temperatures is a prolific source of error. If a temperature-measuring instrument is immersed in a fluid and there are bodies nearby which are at higher or lower temperatures, the probabilities are against the correctness of the temperature measurements with this instrument. For example, if a gaseous material is being heated by allowing it to flow through a tube with walls at a temperature higher than that of the material, a thermocouple immersed in the flowing gas may actually be at a temperature higher than that of the gas because of the heat radiated to it by the hot walls of the tube. Or if a thermocouple is used to measure the temperature of products of combustion flowing through a boiler and is so located that it can "see" a cooler heating surface of the boiler, it will be at a lower temperature than the gases. Experiment shows that radiation to the colder heating surfaces of a boiler may cause a thermocouple to indicate temperatures as much as 150°F. too low when immersed in flue gases at a temperature of about 600°F.

Thermometer Wells.—Under all conditions, methods of temperature measurement which do not involve the use of thermometer wells between the medium and instruments are to be preferred. Unfortunately, they are not always possible under conditions governing the making of power tests. When thermometer wells are used, the following precautions must be observed, particularly when the temperature being measured differs greatly, say, 50°F. or more, from that of the surroundings:

1. The part of the well projecting beyond the container must be as small as possible so as to eliminate heat transfer to or from surroundings.

2. The exposed parts of the thermometer well should be covered with thermal insulating material of a kind appropriate to the temperatures being dealt with. The pipe wall or equivalent should also be well insulated for some distance from the thermometer well if this is not already the case and if the placing of such insulation will not materially affect the conditions of the test.

3. The upper end of a thermometer well if not well insulated, especially if projecting an inch or more beyond the wall of the pipe in which the fluid temperature is desired, may cause an error of as much as 50°F., for inside temperatures of, say, 600°F. and outside of 50°F. Such an error will be caused by the heat loss from the upper end of the well owing to heat transfer to the pipe wall and to the air surrounding the projecting end of the thermometer well.

4. The thermometer well should ordinarily be so constructed that it has the smallest possible metallic section consistent with the necessary strength just inside the wall of the pipe. This will tend to reduce the quantity of heat which can flow along the well with a given temperature difference and, conversely, will tend to maintain the maximum possible difference of temperature between opposite ends of the well.

5. The part of the well exposed to the medium of which the temperature is to be determined must have the necessary surface to permit it to absorb heat as fast as it is

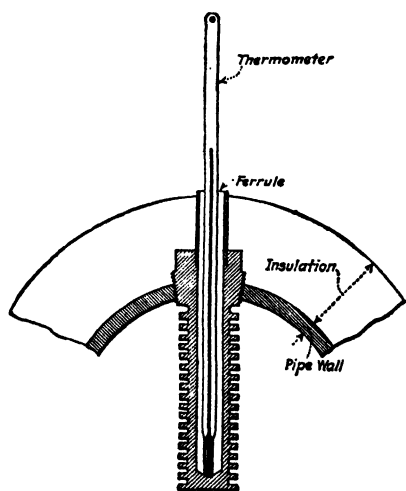


FIG. 37b.—Finned thermometer well.

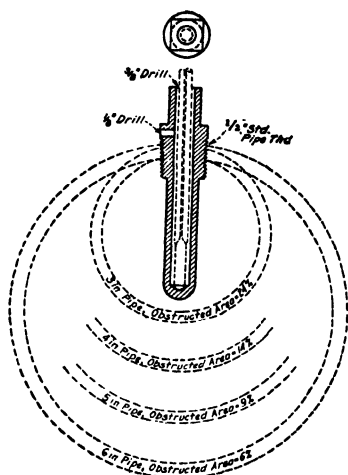


FIG. 38a.—Plain thermometer well and pipe sizes.

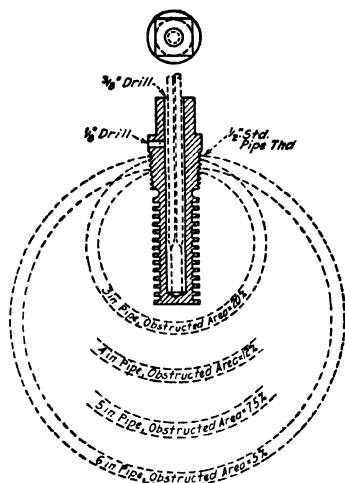


FIG. 38b.—Finned thermometer well and pipe sizes.

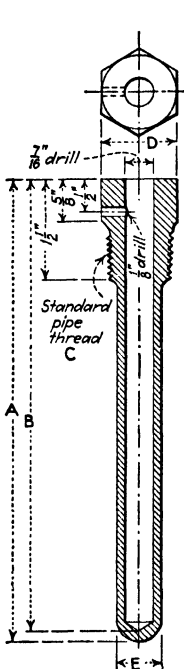


FIG. 38c.—Details of plain thermometer wells for gases and superheated vapors.

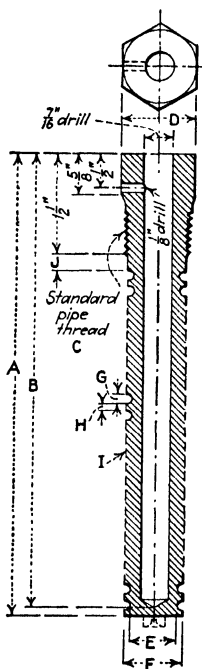


FIG. 38d.—Details of finned thermometer wells for gases and superheated vapors.

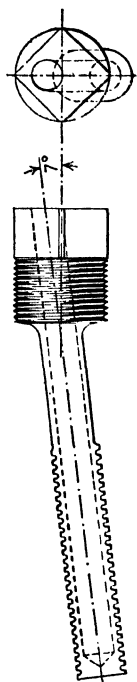


FIG. 38e.—Finned well for vertical pipes.

lost from the exposed end (or to dissipate heat as fast as it is absorbed at the exposed end) without necessitating a temperature difference in excess of that consistent with the degree of accuracy being sought.

When it is necessary to use thermometer wells in measuring the temperature of a fluid within a conduit or vessel, the types to be used are as follows:

Fluid	Type of Well
Liquid	Plain well (Fig. 37a)
Saturated vapor	Plain well (Fig. 37a)
Gas	Finned well (Fig. 37b)
Superheated vapor	Finned well (Fig. 37b)

The fins on the wells used with gas and superheated vapor compensate more or less for the lower rate of heat transfer from the gas to the thermometer bulb within the well, as compared with the rate of heat transfer from a liquid.

Thermometer-well Dimensions.—Plain wells, which are suitable for use in liquids and saturated vapors, are standardized in Table VI. The dimensions refer to Figs. 38a and 38c.

The dimensions of finned wells, which are suitable for use in a horizontal pipe carrying gases and superheated vapors, are standardized in Table VII and refer to Figs. 38b and 38d. A finned well for use in a vertical pipe is shown in Fig. 38e. Dimensions approved by the A.S.M.E. are approximately as given in Table VII.

TABLE VI.—DIMENSIONS FOR PLAIN THERMOMETER WELLS, IN INCHES

Pipe size, inches	A	B	Standard pipe thread	D	E
4 to 7.....	4	3 $\frac{7}{8}$	$\frac{1}{2}$	1	$\frac{1}{2} +$
8 and larger.....	7	6 $\frac{7}{8}$	$\frac{3}{4}$	1 $\frac{1}{8}$	$\frac{5}{8}$

TABLE VII.—DIMENSIONS FOR FINNED THERMOMETER WELLS, IN INCHES

Pipe size, inches	A	B	Standard pipe thread	D	E	F	G	H	No. of fins	J
4 to 7.....	4	3 $\frac{7}{8}$	$\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{3}{32}$	$\frac{1}{16}$	15	$\frac{5}{32}$
8 and larger.....	7	6 $\frac{7}{8}$	$\frac{3}{4}$	1 $\frac{1}{8}$	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{32}$	$\frac{3}{32}$	21	$\frac{1}{4}$

Heat-conducting Material for Filling Thermometer Wells.—Thermometer wells intended for the measurement of the temperature of liquids are usually filled with a liquid, preferably cylinder oil or machinery lubricating oil. The former is suitable also for use with vapors if the

temperature does not exceed 300°F. When, therefore, the temperature to be measured is so low that the oil does not vaporize and when the temperature measured does not fluctuate rapidly, oil may be used. When, however, there are rapid fluctuations in temperature, the relatively large time lag due to the low thermal conductivity of the oil may cause appreciable errors. The most satisfactory materials to be used in steel or iron thermometer wells are given in Table VIII.

TABLE VIII.—FILLING MATERIALS FOR TEMPERATURE RANGES

Temperature Range, Degrees Fahrenheit	Filling Material
Up to 300	Cylinder oil
300 to 500	Mercury
500 to 1,000	Solder ¹
600 to 1,800	Tin ¹

¹ When solder or tin is used, the thermometer should be removed from the well at the end of the test to prevent damage that may be caused by the solidification of the solder or tin.

When the pressure in a duct or pipe is approximately atmospheric and the temperature is not greater than about 475°F., a thermometer may be inserted in a hole made in a rubber stopper (page 267), and then the thermometer and its stopper may be placed in a suitable hole in a duct or pipe for temperature measurement.

When bare (unprotected) thermocouples or resistance thermometers in which the sensitive element is connected to an indicator of some sort by metallic wires or tubes are immersed in material having a temperature greatly different from that of the region through which the metallic connector runs on its way to the indicator, conduction of heat along such metallic parts should be minimized by running them for a sufficient distance through an isothermal zone.

Imagine, for example, that gas is flowing through a flue of circular section as shown in Fig. 39a and that the temperature falls rapidly from the center toward the wall. If a couple is used to measure the temperature of the gas at the center of the flue and is installed so that the leads run radially outward as indicated at *A*, the temperature indicated will be lower than the real temperature for reasons already explained. If, however, the leads run along the center line of the flue for sufficient distance, as shown at *B*, the error will be reduced to a negligible amount. Attention is called to the fact that the same precautions are necessary with respect to any clamps or other fastening devices which are used to hold temperature-measuring devices in place. The instrument should be thermally insulated from the clamps and the latter should have such a shape that they do not conduct any appreciable quantity of heat away from the sensitive element and also that they do not cause a local drop in temperature of the medium in the neighborhood of the element.

Radiation Shielding.—When it is necessary to place the sensitive part of a temperature-measuring device in a position where it can “see” surfaces at materially higher or lower temperatures than that of the medium in which it is immersed, the installation should be so planned as to minimize errors of the kind already described as arising from radiation under such conditions. When the material in which the device is immersed is flowing, this can be done by surrounding it with one or more shields of cylindrical section, so placed that the axes of the cylinders are in the line of flow. Such an arrangement is shown in Fig. 39b. When there is no definite flow, there is

theoretically no means of protection against radiation error. If shields such as those described are used, however, they will afford some protection if arranged in such a way as to prevent the sensitive element from "seeing" the hotter or colder surfaces and so that the material in which they are immersed flows through them by virtue of the convection currents set up by local heating or cooling. Arrangements of this kind should be studied carefully, as they may be so designed as to produce a larger error than the one they are intended to correct.

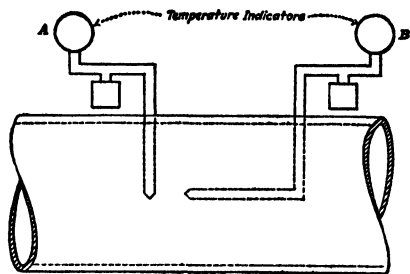


FIG. 39a.—Method of using thermocouple.

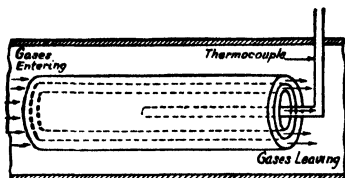


FIG. 39b.—Protecting shields for thermocouple.

Protecting tubes as used with thermocouples are capable of producing errors of great magnitude if improperly used. Such tubes almost always have large external surface, so that they exaggerate radiation errors. When made of metal they have generally large metallic cross sections and are therefore good thermal conductors, so that they cause errors of the type described in connection with thermometer wells. If a thermocouple is merely suspended in such a tube near the closed end, its temperature must of necessity lag behind that of the tube wall, so that, if rapid temperature variations are occurring, the thermocouple is probably never very near to the temperature of the material being measured.

Repairing Thermometers.—Before and after calibrating a thermometer previous to its use, a careful examination should be made of the column of the liquid to ascertain if it is separated. There are four ways in which thermometers may be manipulated in order to rejoin the mercury which has become separated or to remove a gas bubble from the bulb, namely, (1) warming, (2) cooling, (3) tapping on the end, and (4) tapping on the side of the thermometer.¹

Warming the bulb of a thermometer may be done in water or in oil or in the air high above a gas or alcohol flame, but great care must be taken, especially with high-temperature thermometers, not to heat the bulb to a temperature higher than the thermometer is intended to measure. The upper end of a thermometer stem may be warmed over a Bunsen burner if this is done very gradually. When the thermometer is warmed over a flame, it should be rotated about its axis.²

¹ Testing of Thermometers, *Circ.* 8, U. S. Bur. Standards, 2d ed., Sec. 15.

² When a thermometer is installed in a well which is subjected to excessive vibration, breakage of the thermometer can be avoided by placing a small amount of steel wool in the bottom of the well.

Cooling may be accomplished in cold water, ice, or a freezing mixture, but such cooling should not be attempted while the thermometer bulb is still hot to the touch.

Tapping or striking a thermometer on the end must be done carefully at all times, and much more so when the bulb is only partly filled with mercury and the thermometer is inverted. This tapping may be "soft," as when the thermometer is held vertically in the hand and the hand (not the thermometer) is struck on the table, or it may be "sharp," as when the thermometer is held vertically and struck downward on a pad of paper.

Tapping on the side of a thermometer may be done "softly" with the hand or "sharply" with a pencil. Either must be done with care, as there is danger of breaking the stem.

Glass thermometers should always be carried with the stem right side up and excessive vibration or jarring should be avoided. Neglect in this respect may vitiate the results of most careful calibration, as it may cause the mercury column to separate.¹

Figure 40 shows a method of installing the small-size thermometer well in lines smaller than 3-inch. This type of installation should be avoided, however, in steam lines when possible, as thermometers placed near a bend, converging section, or other interrupter of the smooth flow of steam are **apt to read an incorrect temperature because of the throttling and separating effects of the obstruction.** When using this type of installation on small lines (smaller than 3-inch), the tee should be one pipe size larger than the line. When a glass thermometer is used, it should

¹ In some thermometers there is a safety reservoir or "pocket" at the upper end of the capillary tube which serves as a space in which the thermometric liquid can collect in case of overheating, without breaking the bulb by internal pressure. This safety reservoir serves also a useful purpose when reuniting a broken mercury thread as it can be used as a collecting chamber in which some of the mercury from the main bulb can be allowed to flow when the thermometer is inverted.

To remove mercury from the upper reservoir when it is partly full, the following procedure is recommended: (1) Hold thermometer vertically, bulb down. (2) Tap "softly" on side near the top and "softly" on end, to bring mercury to lower part of upper reservoir. (3) Try tapping both "softly" and "sharply" on the end, and, if this fails to bring the mercury down the stem, warm the bulb until the mercury rises and unites with that in the reservoir. (4) Cool very slowly and probably the mercury will all come down. (5) If not, try again, cooling more slowly.

To remove mercury from upper reservoir when this is completely full or when mercury persistently stays at the top of reservoir: (1) Warm the upper end carefully over a Bunsen flame, beginning at the extreme tip. Mercury will be driven down the capillary by its own vapor pressure. (2) If small mercury globules are left, warm a little more and they will condense farther down. (3) Wait until the upper end cools to a temperature comfortable to the hand. (4) Warm the lower bulb until mercury rises and collects the small globules. (5) To remove gas bubbles from high-temperature thermometers see Test Code on Instruments, *Mech. Eng.*, Vol. 48, No. 4, pp. 386-387.

not be lifted out of the thermometer well while the reading is being taken.¹

Care should also be taken to avoid **parallax**, by having the line of sight perpendicular to the stem of the thermometer.

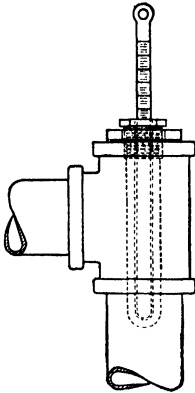


FIG. 40.—Method of putting thermometer well in small pipe.

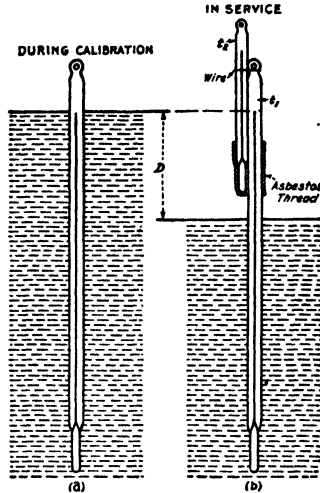


FIG. 41a.—Thermometer calibrated for full immersion and used for partial immersion.

Emergent-stem Corrections.—An “emergent-stem correction” must be made when a thermometer in use has a portion of the emergent stem at a temperature other than that which it had during calibration.² The emergent-stem correction K is to be **added algebraically** to the indicated temperature of a mercury-filled thermometer **before it has been corrected for the errors found during calibration** with proper immersion. For a mercury-filled thermometer calibrated for full immersion and used for partial immersion as shown in Fig. 41a, the stem correction can be calculated from the following formula:

$$K = 0.00009D(t_1 - t_2) \quad (1)$$

¹ An idea of the magnitude of the possible error may be gained from the following: Experiment has shown an error of 30°F. when withdrawing a glass thermometer from a flue containing gas at 600°F. and reading after withdrawal. The same effect in the case of an armored thermometer gave an error of about 13°F. due to the fact that the armor contained sufficient heat materially to decrease the rate at which the liquid in the thermometer cooled.

² The stem correction can be reduced by surrounding the thermometer with a glass tube about $\frac{3}{4}$ inch in diameter. This produces a region around the thermometer much hotter than the surrounding atmosphere. The auxiliary and main thermometers are easily read through the glass tube. When the thermometer is used in a well, the glass tube can rest on the top of the thermometer well.

where K = correction, in degrees Fahrenheit.

D = emergent stem, which is the length of exposed mercury thread, in degrees Fahrenheit on thermometer scale.

t_1 = temperature indicated by the thermometer, in degrees Fahrenheit.¹

t_2 = temperature indicated by an auxiliary thermometer having its bulb placed about three-quarters down the exposed mercury thread as shown in Fig. 41a, in degrees Fahrenheit.

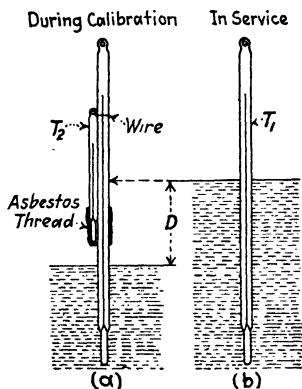


FIG. 41b.—Thermometer calibrated for partial immersion and used for excessive immersion.

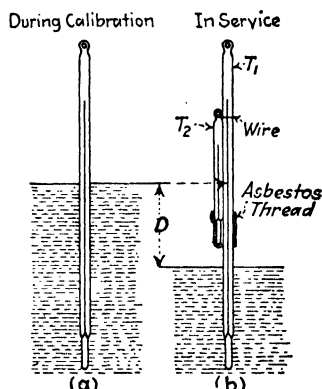


FIG. 41c.—Thermometer calibrated for partial immersion and used for deficient immersion.

This equation, it will be observed, includes three factors: coefficient of expansion, length, and temperature difference. The coefficient of expansion given in the equation is the difference between the volumetric coefficient of expansion of mercury (0.00010) and the linear coefficient (0.00001) of the kind of glass ordinarily used for thermometers.

The correction K for various values of D , t_1 , and t_2 is given in the chart (Fig. 42). This correction is to be **added** to the observed temperature in the usual cases when t_1 is greater than t_2 and subtracted when, as in measuring the temperature of refrigerants, t_1 is less than t_2 . It should be noted that there are two scales of corrections in this figure. The left-hand vertical scale is the correction for values of $t_1 - t_2$ from 0 to 250°F., and the right-hand scale from 250 to 1000°F.

Example.—The observed thermometer reading is 500°, temperature of the stem is 70°, and immersion is up to 300°. All temperatures Fahrenheit. Stem correction by formula (1) is

¹ Inasmuch as t_1 is not the true temperature of the bulb of the immersed thermometer, the correction K is only approximate upon the first substitution in the foregoing formula. If a new substitution in the formula is made using $t_1 + K$ as the new value for t_1 , the new correction K will be nearer correct than the first value. Further recalculation with t_1 corrected for the new value of K will result in a still closer value for K . Seldom are more than two recalculations necessary, and then only for high temperatures and long-exposed stems. Long emergent stems should always be avoided.

$$K = 0.00009 \times 200(500 - 70) = +7.57^{\circ}.$$

Corrected reading is $500 + 7.57$, or nearly 507.6°F . As read from the curves in Fig. 42a, the correction is $+7.6^{\circ}\text{F}$.

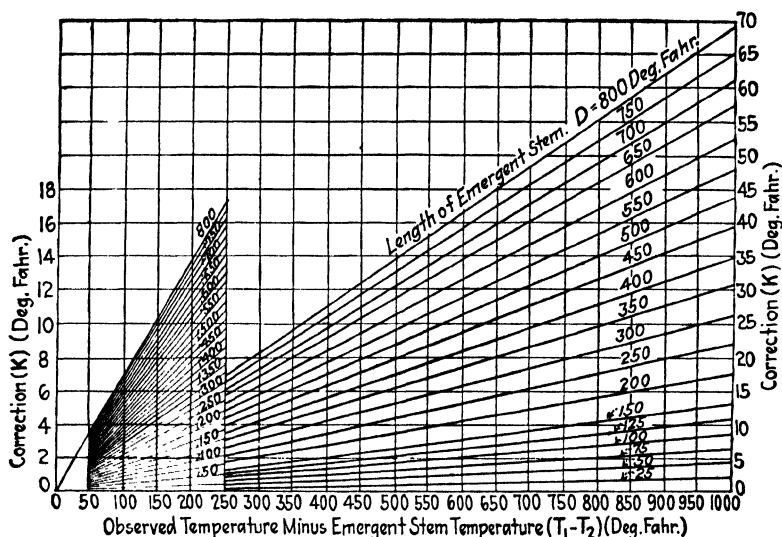


FIG. 42.—Exposure corrections for engraved stem thermometers.

NOTE.—The left-hand vertical scale of corrections (ordinates) is for values of $t_1 - t_2$ (abscissas) from 0° to 250° , and the right-hand scale is for values from 250° to 1000° .

When the **partial-immersion thermometer** is used with an excessive immersion (see Fig. 41b), it is assumed that the part of the mercury thread which was emergent during the calibration, but which is now immersed, is at the same temperature as the bulb of the thermometer, which is approximately the same as the indicated temperature. The stem correction is calculated from the following formula:

$$K = 0.00009 D (t_1 - t_2) \quad (1')$$

where K = correction, in degrees Fahrenheit.

D = excessive immersion of thermometer, in degrees Fahrenheit.

t_1 = temperature indicated by thermometer, in degrees Fahrenheit.

t_2 = average temperature of that part of stem represented by D during calibration of thermometer with proper immersion, in degrees Fahrenheit.

The correction K for a partially immersed thermometer can be obtained from the chart given in Fig. 42. It should be subtracted if t_1 is greater than t_2 , or it should be added if t_1 is less than t_2 .¹

¹ Thermometer corrections to be applied to other cases of variation of depth of immersion and external temperature, as, for example, in Fig. 41c, are given in

In making a correction for the excessive immersion of a partial-immersion thermometer by the foregoing method, it is assumed that the thermometer has already been calibrated for the proper immersion and for the same external temperature as exists during use. During the calibration the average temperature of the emergent mercury thread, expressed in degrees, should be measured by an auxiliary thermometer three-quarters down the exposed section as shown in Fig. 41b.

After the stem correction K is applied to the indicated temperature, the correction determined from the thermometer calibration must also be made to obtain the correct temperature.

External-pressure Correction.—The external pressure to which the bulb of a thermometer is subjected affects the indication of the instrument and wide variations in pressure cause very significant errors. Thermometers are ordinarily calibrated at or very nearly at atmospheric pressure, and if used under very different conditions the possible magnitude of the error should be determined.

The extent to which variations in pressure will modify the indications of the thermometer varies with the thickness of the glass of the bulb. Those thermometers built to have the smallest time lag also have, in general, the thinnest bulb wall, and thus the better they are in this respect the more apt they are to require a correction for pressure.

Pressure corrections are generally ignored in the belief that they are of such small magnitude as to be negligible under the worst conditions. This is not necessarily true and should not be assumed to be true without investigation. High-grade engraved-stem thermometers have bulbs with wall thickness between 0.5 and 0.7 millimeter and for such thermometers the correction for external pressure is of the order of 0.01°F . per pound per square inch. Such a thermometer immersed in a nearly perfect vacuum would read about 0.14°F . too low. On the other hand, such a thermometer projecting through the wall of a tank into a deep body of liquid may, under certain conditions, read too high by several times 0.14°F .

Industrial thermometers generally have bulbs sufficiently heavy to withstand variations of pressure to which they may be expected to be subjected, but this should be checked in cases of doubt.

To determine the external-pressure correction for a thermometer, the change in position of the mercury column is measured when the thermometer is subjected to different external pressures. It is most convenient to use atmospheric pressure as one pressure and a partial

vacuum for the other. The apparatus employed in checking an engraved-stem thermometer is a glass tube into which the thermometer can be inserted, and which may be connected by means of a two-way cock to the atmosphere or to a vacuum-producing apparatus. A mercury manometer is used to measure the pressure to which the thermometer is exposed. The glass enclosing the thermometer should contain some mercury at its lower end in which the bulb of the thermometer is immersed, to enable it to take up the surrounding temperatures readily, and above this pure glycerine should be introduced to facilitate the reading of the thermometer through the glass tube. The whole of the tube should be enclosed in a bath of water, the temperature of which can be controlled. Readings of the thermometer are taken with the enclosing glass tube alternately under atmospheric and reduced pressure. The readings at reduced pressures should be taken rapidly so as to prevent the possible cooling of the glycerine due to evaporation of the associated alcohol. The external pressure coefficient B in **degrees Fahrenheit per pound per square inch** change in pressure is given by the relation

$$B = \frac{t_1 - t_2}{p_1 - p_2}, \quad (2)$$

where t_1 = reading of thermometer, in degrees Fahrenheit, when exposed to pressure p_1 .

t_2 = reading of thermometer, in degrees Fahrenheit, when exposed to pressure p_2 .

p_1, p_2 = different external absolute pressures, in pounds per square inch, to which the thermometer is exposed.

For convenience in applying the pressure coefficient to a thermometer requiring this correction, it is desirable to calculate a table giving values of the correction for each pound-per-square-inch change in the external pressure to which the thermometer is likely to be subjected.

Calibration of Thermometers.—The calibration of glass thermometers can be made in several different ways, namely, (1) by comparison, in thermometer-comparison baths in which the temperature can be controlled, with thermometers that have been calibrated by the U. S. Bureau of Standards at Washington, D. C.;¹ (2) by comparison with substances having known melting or boiling temperatures; (3) by checking the ice-point and the boiling-point-of-water readings and then checking the uniformity of bore and the uniformity of the scale² at other parts of the scale, or (4) by comparison with steam of known temperature, such as

¹ In England, such calibrations will be made by the National Physical Testing Laboratories, London; and in Germany by the Reichsanstalt, Berlin.

² As the checking of the uniformity of the bore and the scale is a tedious and very painstaking job, it will not be discussed here. Those interested in the methods used are referred to Edser's "Heat for Advanced Students," 1st ed., pp. 29-34, The Macmillan Company.

the saturation temperatures corresponding to known pressures. The checking of the ice-point and boiling-point-of-water readings are recommended for the checking¹ of standard thermometers to see if their calibrations are shifting.

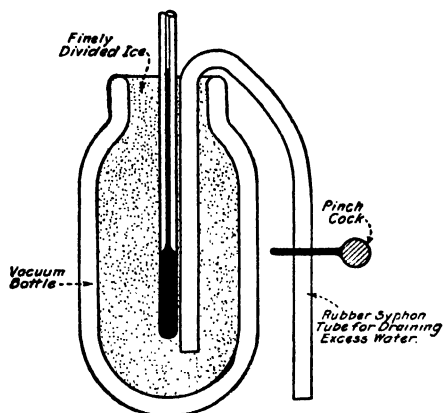


FIG. 43a.—Apparatus for ice-point test.

distilled water is used. Figure 43a shows an approved form of apparatus for determining the ice point.

Boiling Point.—To test the thermometer at the boiling point of water, that is, to check the “boiling point,” suspend the thermometer so that it will be entirely surrounded by the vapor of boiling distilled water at atmospheric pressure. The boiling should be continued until the water is free of air, which will be indicated by the temperature becoming constant, assuming, of course, that the barometric pressure has not changed. Care must be taken to make certain that the thermometer does not project into the liquid. Note the reading. From the barometric pressure and the Steam Tables determine the boiling point for atmospheric pressure. The difference between the reading of the thermometer and the boiling point for atmospheric pressure will be the error in the marking of the boiling point. Figure 43b shows an approved form of apparatus for determining the boiling point.²

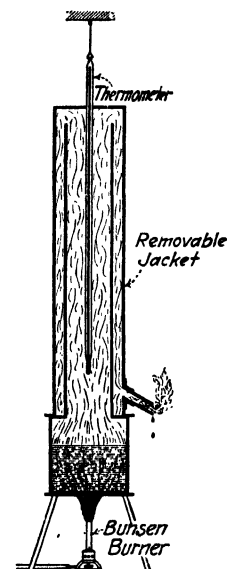


FIG. 43b.—Apparatus for boiling point of liquids.

¹ It is not permissible to assume that the indications over the whole scale have shifted an amount equal to that at the ice point. A standard thermometer, of which the ice point has shifted, cannot be used with certainty without a recalibration by the U. S. Bureau of Standards.

² A more complete description of the methods used in ice-point and boiling-point determinations will be found in the U. S. Bur. Standards *Reprint* 69, On the Standard Scale of Temperature in the Interval 0° to 100°C., by Weidner and Dickinson.

Calibration by Comparison with a Standard or Secondary Standard Thermometer.—In this method of calibration the thermometer which is to be calibrated is immersed along with the standard thermometer in a suitably designed vessel. Baths used in the calibration of thermometers by comparison with standards differ in attainable temperature

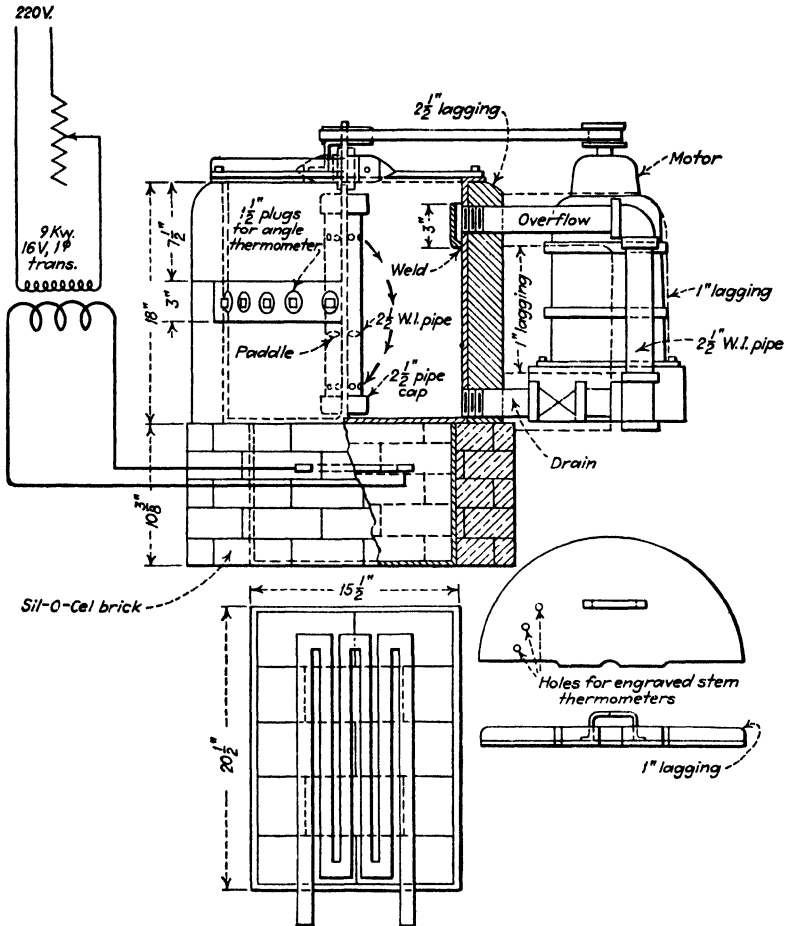


FIG. 44.—Thermometer comparison oil bath.

range and also in construction. The essential feature of a thermometer-comparison bath is uniformity of temperature throughout the part of the bath in which the standard thermometer and thermometer to be checked are located. Obviously, the water-bath type cannot be used at temperatures above the boiling point of water at atmospheric pressure.

Figure 44 shows an oil-bath equipment recommended by the A.S.M.E. Power Test Committee. In the side of the tank and about 9 inches from

the top, several holes are drilled, tapped, and plugged with pipe plugs. These holes are for the insertion of industrial-type angle thermometers when calibration of this kind of thermometer is desired.

The stirring mechanism consists of a small propeller mounted on a vertical shaft, the propeller being enclosed in a $2\frac{1}{2}$ -inch pipe with caps at the top and bottom. The propeller for stirring is driven by a vertical shaft electric motor. In the operation of the apparatus, the propeller draws oil through the holes at the bottom and pumps it out through holes at the top of the $2\frac{1}{2}$ -inch pipe, thus producing a good circulation of oil.

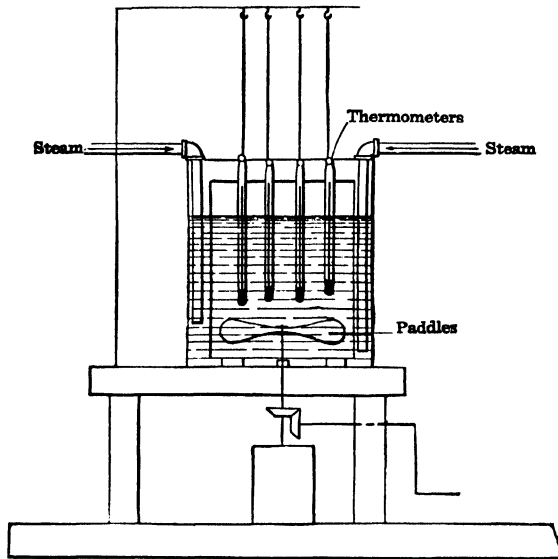


FIG. 45.—Simple apparatus for calibration of thermometers at temperatures less than the boiling point of water.

Lagging insulation is provided on the outside of the tank and around the overflow and drain pipes.

The oil bath is heated electrically by a resistance heater located in a sheet-iron box below the calibrating tank. Electric power is supplied to the heater through a transformer which is designed for a suitably low secondary voltage. Rheostats in the primary circuit of the transformer make it possible to obtain practically any temperature between that of the room and the vaporizing point of the oil that is used. It is desirable to insert, when the apparatus is built, a thermocouple very close to the electric heater so that its temperature can be observed and thus avoid injuring it by overheating.

For temperatures between 200 and 600°F. oil is the most satisfactory liquid for filling a comparison bath. Mineral oils with high flash points or the cooking compound Crisco may be used; all such materials at present available gradually decompose and char.

In calibrating thermometers by the use of molten metals, a correction for the **pressure of the molten metal** should be applied to thermometers which are to be used with less external pressure. The method of making this correction is explained on page 48.

The melting points given are for "pure" metals. Chemically pure metals which are obtainable from any reliable chemical-supply house are not absolutely pure, and their melting points may vary slightly from those given above. In case of doubt the U. S. Bureau of Standards certified samples are recommended.

These metals can be melted in iron crucibles heated with gasoline blowtorches. The crucible should be surrounded with fire brick, magnesia blocks, or some other heat-insulating material to prevent rapid loss of heat; otherwise it might not be possible to obtain temperatures high enough to melt zinc and lead. Suitable crucibles can be made of pieces of clean 1-inch black-iron pipe 8 inches long and closed at one end with caps. It is advisable to make separate crucibles for each metal, and to keep the metal in its crucible after the calibration is complete. The use of separate crucibles will tend to prevent possible contamination of the metals and avoid the difficulties that will surely be encountered in taking the metals out of the crucibles and putting them in again when needed in future test work. All contamination should be carefully guarded against, as the melting points of the metals would be lowered, and serious errors might result. As a matter of fact, the metals are contaminated to a slight degree by being heated in the iron crucibles for a long period. This contamination seldom lowers the melting points more than 1°F.

When the metal is heated, the exposed surface should be covered with graphite or powdered charcoal to prevent rapid oxidation. The depth of the molten-metal bath should be such that the insertion of the thermometer will bring the top surface of the graphite to the top of the crucible. The metal should first be melted and then heated about 50°F. above its melting point. Then the thermometer should be inserted to the proper depth of immersion. The blowtorches should then be removed, and, while the metal is cooling, readings should be taken every 10 seconds until the metal has solidified, which takes 5 to 10 minutes. The readings will show at first a marked drop in temperature; but *when the metal begins to solidify*, the temperature will remain constant for a number of readings. After the metal has solidified, there will again be a uniform drop in temperature. The constant-temperature readings are the freezing point of the metal. The thermometer should be taken out of the metal bath as soon as the latter reaches a spongy or granular consistency. Otherwise the contraction of the metal is likely to crush the thermometer. When the temperature readings are plotted as ordinates and the time as abscissas, curves are obtained which have a decidedly flat part. The flat part of the curve is the melting temperature of the metal and is one of the points for the calibration curve.

Calibration by Reference to Substances of Known Boiling Points.—

When one is calibrating by reference to substances of known boiling points, the thermometers are immersed in the vapors of the boiling substances. The materials most commonly used are given, with their boiling points, in Table X. In all cases chemically pure materials are referred to.

TABLE X.—BOILING POINT OF REFERENCE SUBSTANCES

Substance	Degrees Fahrenheit
Naphthalene.....	424
Benzophenone.....	582
Anthracene.....	644
Sulphur.....	832

The temperatures given for boiling points are for pressures of 29.92 inches of mercury absolute. For other pressures p in inches of mercury absolute, the corresponding boiling temperature t in degrees Fahrenheit may be obtained with sufficient accuracy for small differences of pressure, by using the following formula:

$$t = t_0 + A(p - 29.92),$$

where t_0 is the boiling temperature in degrees Fahrenheit at 29.92 inches of mercury as tabulated above, and A is the correction factor given in Table XI.

TABLE XI.—BOILING-POINT CORRECTION FACTOR

Substance	Correction Factor
Naphthalene.....	2.61
Benzophenone.....	2.88
Anthracene.....	3
Sulphur.....	4.21

The boiling points can be readily determined in the standard Meyer tube apparatus shown in Fig. 46. The bulb of the tube can be heated with a gas flame. The asbestos or aluminum cone surrounding the bulb of the thermometer prevents condensed vapor from running down over the bulb, thereby cooling it below the temperature of the surrounding vapor, and it eliminates direct radiation from the bulb to the colder walls of the large tube. An unprotected thermometer would read in the order of 1°F. low in the vapors of a sulphur bath.

Calibration by Reference to Temperatures of Saturated Steam at Known Pressures.—There is a definite temperature of saturated steam corresponding to every pressure. If, then, the absolute pressure is known, the temperature corresponding to it can be obtained from the Steam Tables in the Appendix.

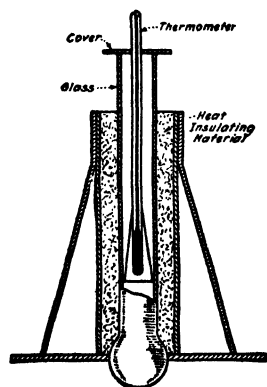


FIG. 46.—Apparatus for boiling point of solids.

Thermometers to be calibrated are placed in adjacent mercury-filled thermometer wells (T , T , Fig. 47) located in a cylindrical steam drum. Water must be kept in the steam drum to insure against possible superheat of the steam. Pipe connections are provided for an accurately calibrated pressure gage. Since a pressure gage is calibrated by a dead-weight apparatus, it is better to determine the steam pressure directly by the dead-weight apparatus and thus eliminate a possible error. A mercury column may be used for calibration at pressures which fall within its limits. A , B , and E are valves for regulating the

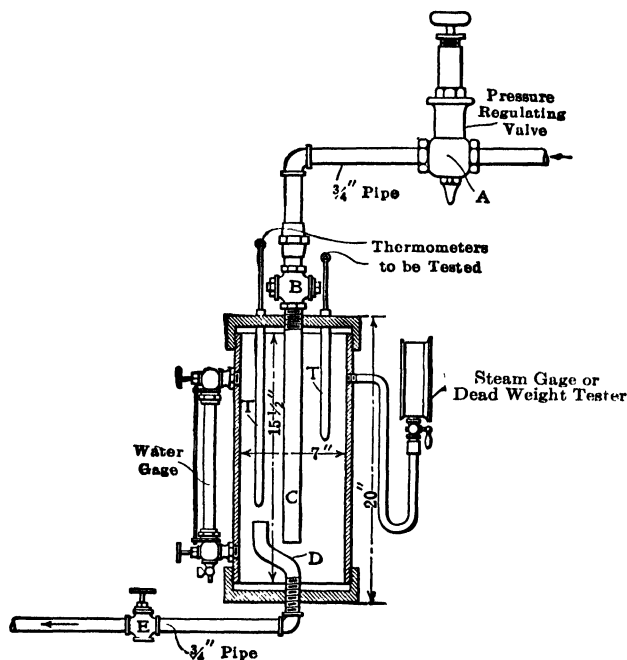


Fig. 47.—Apparatus for calibration of thermometers with saturated steam.

flow of steam through the drum. Figure 46 does not show a barometer, which is essential. Attention is called to the fact that errors of several degrees Fahrenheit may be expected unless the equipment is well lagged, a reasonable flow of steam is maintained, and, above all, the pressure is accurately determined.

The steam passes through the regulating valve A and enters the large cylindrical drum through the pipe C , which is open at the lower end. Exhaust is through D . A water gage is provided to show the level of water. Obviously, if the bottom of the long thermometer well is immersed in water, the temperature indicated would not be comparable, as a rule, with the shorter thermometer exposed entirely to steam. The top of the exhaust pipe is made higher than the bottom of C so that there

will always be some water "trapped" in the cylinder; and the steam which enters must bubble up through this water, making it more certain that the steam to which the thermometers are exposed is wet. In other words, the superheating which is likely to be caused by throttling with the regulating valve *A* must be eliminated. Although only two thermometers are shown, the apparatus can be very conveniently fitted with four or six thermometer wells so that a greater number can be calibrated at a time. The thermometer wells may be filled with cylinder oil or, preferably, for high temperatures, with mercury.¹ Temperature is varied by throttling with the valves on either or both the steam inlet and discharge pipes. Usually the necessary adjustment is made more easily by manipulating the discharge valve rather than the inlet. At least **5 minutes** should be allowed after the valves have been adjusted for the mercury in the thermometers to come to rest before readings for comparison are taken. Readings of both the standard and the thermometer being calibrated must be taken as nearly as possible at the same time, and the thermometers should be lifted from their wells, when necessary, just enough to bring the mercury into view. Observations should be taken as quickly as possible to avoid errors due to cooling and should be made with approximately the same increments.

The method of calibrating thermometers for temperatures between 212 and 400°F. by comparison with steam tables is more generally used than any other method.

It is very important that the thermometer being calibrated should be immersed in the well to the same extent as it will be when in use. It is a very good arrangement to have a standard type of well for all thermometers of the same length, so that, if a thermometer has been calibrated in one of these standard wells, the calibration will be applicable in any one of the wells, provided the "room" temperature does not vary widely. Effect of variations of "room" temperatures (emergent-stem correction) can be readily calculated by the method explained on pages 45 to 48.

Another device used to change superheated steam to the saturated condition is illustrated in Fig. 48. In principle it is the same as Fig. 47. In this apparatus the steam passes down through the vertical supply pipe *S*, closed at the lower end, and escapes from perforations near the bottom to bubble up through the water contained in the chamber *A* and is carried away in the pipe *D*. In this way the steam can be made to lose enough

¹ If oil is used in thermometer wells, precautions should be taken that the oil is absolutely free from water and that there is no water in the wells. If water is allowed to accumulate, there may be a slight explosion, sometimes strong enough to throw the thermometers out of the wells.

Thermometers used for measuring high temperatures should be protected from contact with the metal of the wells by wrapping the stems at the mouth of the well with cotton waste or with rings of cork.

heat to the water to reduce a reasonable amount of superheat. No valves or other devices that have a tendency to throttle the steam and consequently superheat it should be placed between the saturating drum and the steam drum in which the thermometers are to be calibrated.

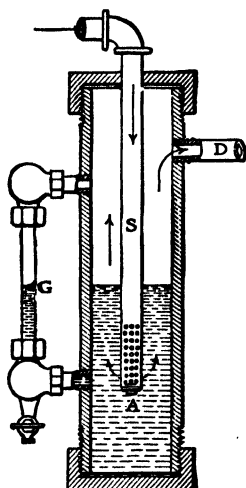


FIG. 48.—Device to reduce the superheat in steam.

A most excellent method for making calibrations of thermometers by means of a steam drum is a combination method. That is, the corrections are calculated from the temperature corresponding to the corrected gage pressure¹ and are at the same time checked by comparison with a "standard" thermometer.

In all experimental work in engineering, such methods of checking results cannot be too highly commended. Checking not only assists in the elimination of errors of observation as well as in calculations but for the engineer it is his key to success.²

The observations should be tabulated in the form given on page 59. The column under the heading "Standard thermometer" is to be left blank, of course, when only the steam gage is used for the calibration. To obtain the **absolute pressure** the barometric pressure must be observed during the calibration.

Curves.—1. Plot a curve for each thermometer showing observed temperatures of the thermometer tested (abscissas) and the corresponding "true" or "standard" temperatures (ordinates). This curve is of no value, however, unless a reasonably large scale is used.

2. Plot an **error curve**, taking observed temperatures of thermometer tested for abscissas and "errors" for ordinates. Compare with Fig. 31.

Recording Thermometers.—Recently instruments for recording automatically low as well as high temperatures have been very satisfactorily developed. There is a sensitive bulb suitable for being inserted into a pipe fitting or duct and attached by a capillary connecting tube to the recording instrument.

¹ Since thermometers are calibrated usually only for increasing temperatures, the gage corrections to be applied should be those corresponding to increasing pressures. If only the average corrections of the gage are available for comparison, then the gage should be tapped lightly on the back of its casing when each observation is taken. Jarring has the effect of eliminating to some extent the lagging effects due to friction, which are presumably eliminated entirely when readings are taken with both increasing and decreasing pressures.

² The **plotting of curves** is still another means which the engineer uses continually for checking his calibrations, his tests, and his conclusions. It has been well said that "the physician buries his mistakes but that the mistakes of the engineer bury him."

CALIBRATION OF THERMOMETER BY COMPARISON WITH TEMPERATURES CORRESPONDING TO STEAM PRESSURES

Record:

1. Date and names of observers.....
2. No. and type of gage.....
3. No. of standard thermometer.....
4. Identification of thermometer tested.....
5. Limits of graduation of both.....
6. Barometer reading.....inches mercury = ...pounds per square inch...

1	2	3	4	5	6	7	8	9	10	
No. of reading	Time of reading	Observed temperatures		Pressures, pounds per square inch				True temperature from (8) and steam table, degrees Fahrenheit	Error of thermometer tested compared with (9), degrees Fahrenheit ¹	Remarks
		Standard thermometer	Thermometer tested	Gage readings	Barometric	Corrected gage readings	Absolute (6) + (7)			

¹ When the sign is +, the correction ("error") should be added, and when -, should be subtracted from the observed reading.

Bourdon Tube Thermometers.—There are three kinds of Bourdon tube thermometers, classified according to the kind of fluid with which they are filled. In class 1¹ are those of the liquid-filled kind, the liquid filling completely the bulb, capillary tube, and the spring that actuates the indicating mechanism of the thermometer (Fig. 49). In this case, liquid expansion is the actuating medium. Those in class 2 are only partly filled with liquid, most of which is in the bulb, and the vapor of the liquid fills the capillary tube and the spring of the indicating device (Fig. 50). In this case vapor pressure is the actuating medium. Class 3 is filled completely with gas. The liquid-filled kind (class 1) and the gas-filled kind (class 3) depend for their operation on the expansion of liquid and gas, respectively. The vapor-pressure kind (class 2) is operated by the pressure inside the spring of the indicating mechanism; this pressure

¹ The classification given here is that used in the Power Test Codes of the A.S.M.E.

depends entirely on the temperature of the free surface of the liquid in the bulb.

Advantages and Disadvantages of Class 1 Bourdon Tube Thermometers.—There is one advantage that a liquid-filled class 1 Bourdon tube thermometer has over the gas-filled kind (class 3): it is that it can be made with a smaller bulb than the gas-filled kind. In this respect, however, it has no advantage over the vapor-pressure kind (class 2). The liquid-filled kind has, however, the important disadvantage that there may be large temperature errors if the capillary tube is long. The length of the capillary tube for accurate results should not exceed about 10 feet and should certainly not exceed 25 feet unless temperature effects

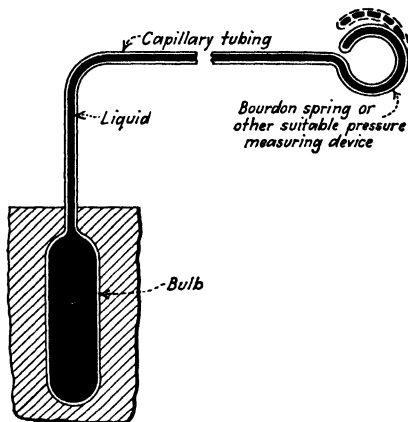


FIG. 49.—Bourdon-tube type completely filled with liquid.

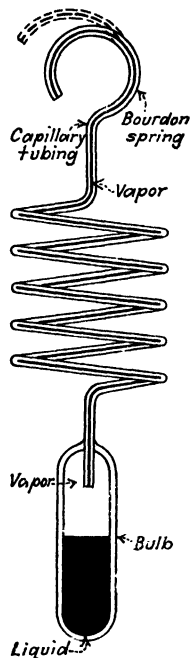


FIG. 50.—Bourdon-tube type operated by vapor pressure.

along the tube are compensated by suitable means.

Advantages and Disadvantages of Class 2 Bourdon Tube Thermometers.—If the capillary tubing of a vapor-filled Bourdon tube thermometer contains only vapor, the temperature indications of the instrument are not subject to error due to variation of the temperature in the tubing and the spring. If, however, the spring that actuates the indicating mechanism, or a part of the capillary tubing, is filled with liquid, an error from fluctuation of temperature around the spring or along the tubing may result if the bulb temperature is higher than the spring or tubing temperature. In general, however, a vapor-filled Bourdon tube thermometer is not subject to error due to variation of temperature of the capillary tube when the instrument is properly set up to avoid the presence of any of the liquid in other parts than the bulb. This kind of Bourdon tube thermometer has the important

advantage over the liquid-filled kind that it can be used for any length of tubing up to 200 feet. Its principal disadvantage, however, is that the pointer of the indicating device does not move through equal increments in all parts of the scale that it covers and, therefore, the charts of most instruments of this kind are nonuniform as to the scale, and therefore average values cannot be obtained with an ordinary planimeter. Another disadvantage is that the accuracy of this kind of instrument is affected by elevation of the bulb above or below the spring actuating the indicating mechanism. When a vapor-filled Bourdon tube thermometer is provided with a non-rigid bulb known as the "capillary bulb" (Fig. 51), the indicating mechanism measures the temperature of the *hottest point* along the length of this relatively long bulb.

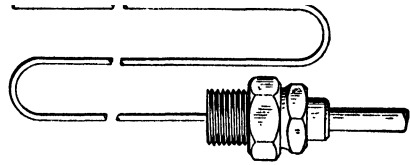


FIG. 51.—Capillary bulb.

Advantages and Disadvantages of Class 3 Bourdon Tube Thermometers.—The component parts of a gas-filled Bourdon tube thermometer (class 3) are almost identical with those of the liquid-filled kind (class 1) except that a different fluid is used. Like the liquid-filled kind, it operates by the thermal expansion of the fluid in the bulb. In most instruments of this kind this expansion fluid is nitrogen, which has a very large working range, between -60 and 1000°F . The chief advantage of this kind of instrument is that there are no restrictions as to the length of the capillary tubing that may be used. In fact such thermometers can be used successfully in straight runs with several hundred feet of capillary tubing. The principal disadvantage of this kind of instrument is that its indications will be in error when the temperature of the capillary tubing is very different from the prevailing temperature when the instrument was calibrated. In case, however, the volume of the gas in the bulb is relatively large in comparison with the volume in the capillary tubing, errors of this kind may be negligible.

Gas-filled Bourdon tube thermometers are often made with a nonrigid bulb which is called a "capillary bulb" (Fig. 51). When the instrument is provided with a bulb of this kind, it measures the *average temperature* along the entire length of the bulb. The limits of error in the ordinary use of the three classes of Bourdon tube thermometers vary from 2 to 10°F .¹ Bourdon tube thermometers can be used to measure the tempera-

¹ The results obtained with vapor-pressure thermometers may be so much affected by the arrangement of the installation, due in most cases mainly to hydrostatic pressure in the bulb and the tubing and to the way the instrument is carried and handled during installation, that special consideration should be given to location when accuracy warranting calibration is expected. Eight methods of installation of vapor-filled Bourdon tube thermometers are discussed in A.S.M.E. Power Test Codes, Part III, Chap. VI, pp. 30-33, inclusive.

ture of fluids within pipes by connecting the bulb into pipe fittings somewhat as shown in Fig. 52.

Calibration of Bourdon Tube Thermometers.—In calibrating thermometers of the Bourdon tube type, the bulb and the indicating mechanism should be at the same relative elevations and also the bulb should be immersed to the same depth as when in use. During the progress of the calibration, the capillary tubing of a liquid-filled (class 1) or gas-filled (class 3) thermometer should be at as near the average temperature it will have in service as is possible. On the other hand, vapor-filled (class 2) thermometers should have the air pressure surrounding the instrument as nearly as possible the same

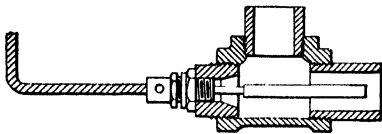


FIG. 52.—Thermometer bulb in pipe fitting for temperature measurement.

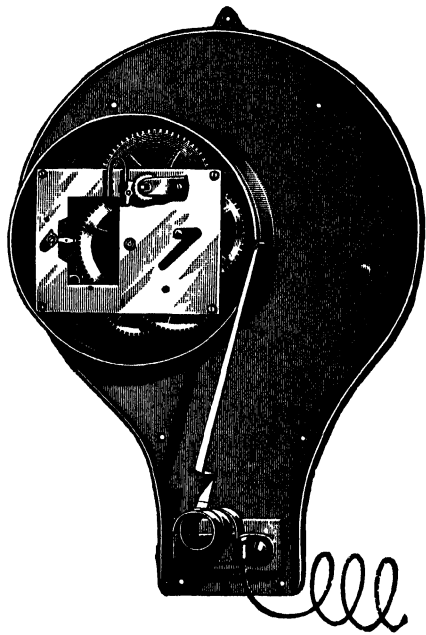


FIG. 53.—Mechanism of a recording thermometer.

as it will be in average use. This precaution applies particularly to service conditions at high elevations.

Methods of calibration of Bourdon tube thermometers are, of course, in essential features, the same as for other kinds of thermometers as explained on pages 49 to 59.

One of these instruments is shown in Fig. 53 with the cover removed so that the mechanism can be seen. It is exactly the same as that of a recording pressure gage (see page 19).

In general appearance and in the operation of the recording mechanism and clockwork, these recording thermometers are like the recording pressure gages now in general use. Some of these recording instruments have a short rigid connection between the bulb and the recording mechanism, which makes it necessary to locate the instrument always immediately adjacent to the bulb. In Fig. 53 there is a flexible connection of capillary tubing attached to the bulb permitting the setting up of the instrument on a wall near by. This capillary tube must, however, be handled very carefully to prevent causing a serious leak, making the instrument useless.

Pyrometers.—One usually thinks of a thermometer as suitable for the measurement of temperatures up to about 600°F ., because of the common use of mercury-in-glass thermometers up to that limit. An instrument used for measuring higher temperatures is generally called a *pyrometer*. Nevertheless, there are a number of types of temperature-measuring instruments that cover a very wide range as shown in Table V (page 38). For example, electric-resistance thermometers may cover a range between -400 and 1800°F .

Thermoelectric Pyrometers.—When two wires of different metals are joined at both ends, so as to form a complete metallic circuit, as in Fig. 54, and if the two junctions *H* (hot) and *C* (cold) are at different temperatures, an electromotive force is generated which can be measured to an arbitrary scale with a galvanometer or commercial millivoltmeter in series with the two wires. If the cold junction is always maintained at a constant temperature, the scale of the galvanometer can be graduated to read directly the temperature of the **hot junction**.

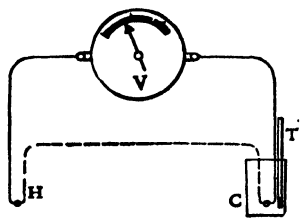


FIG. 54.—Diagram illustrating action of a thermocouple.

In practice it is usually impracticable to maintain a constant temperature at the cold junction, so that usually a *compensating device* is arranged to eliminate the error. One of these devices (Fig. 55) consists of an air-tight glass bulb partly filled with mercury into the top of which a U-shaped platinum loop is fused. This platinum loop is long enough to extend into the mercury and its ends are connected to be in series with the thermocouple at the cold junction. When the temperature of the leads of the outside circuit falls, the voltage due to the couple increases because of the greater range of temperature at the “hot junction,” but the mercury in the bulb contracts so that the current must pass through a greater length of the high-resistance platinum wire in the loop. The net effect is that the increased resistance neutralizes the greater voltage produced at the “hot junction.” Another method of compensation is to attach a small mercury thermometer to the cold couple and put a wire resistance in series with the circuit.

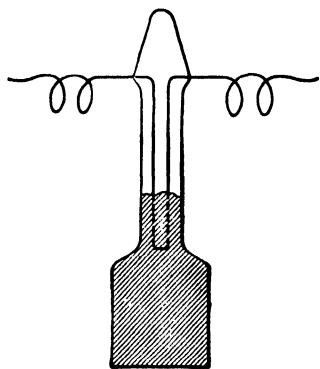


FIG. 55.—Mercury compensating device for thermocouples.

There are two general types of such instruments: (1) high resistance, (2) low resistance. The high-resistance type has a *couple* formed of two wires of small diameter. One wire is platinum and the other is an alloy of 90 per cent platinum and 10 per cent rhodium.¹ To protect the fine and delicate wires against breakage and also because platinum deteriorates

¹ Formerly iridium was used for alloying but it volatilizes rapidly at over 1500°F ., causing a gradual lowering of the voltage produced.

in the silicon, phosphorus, and other "gases of reaction," the couples of the high-resistance type are always protected by porcelain or iron tubes. If the temperature never much exceeds 1500°F., iron tubes are satisfactory. For higher temperatures porcelain tubes are used, but they must be handled carefully as they are easily cracked. Base-metal or low-resistance types of couples are usually made of alloys of nickel, iron, and copper. Couples used very largely in America for temperatures up to about 1500°F. are made of nickel-steel and copper; another is made of one wire of nickel and the other of an alloy of nickel and chromium. Such couples made of cheap metals can be made of larger rods and at less cost than the high-resistance type. Even though they are not quite so accurate, they are generally preferred for industrial work. There is also the advantage that a broken couple can be readily replaced by rods of **iron and copper** fused together at one end. A calibration curve for the couple is easily made to be accurate enough for practical purposes. The principal consideration in selecting rods for such couples is to get them of uniform chemical composition, and they should be annealed, preferably in an electric furnace, to a temperature higher than that to which they are to be exposed. If rods in a couple are not of uniform composition, parasitic currents are produced which oppose that produced by the couple at the junction. Since these wires can be made comparatively large, usually about $\frac{3}{8}$ -inch diameter, the current generated will be large compared with the high-resistance types and its change in resistance with change in temperature will be small, so that a cheaper low-resistance galvanometer can be used. Low-resistance couples are usually protected by an iron tube, mainly because the steel wires deteriorate in the presence of sulphur gases and the asbestos insulation needed for separating the rods along their full lengths is likely to last longer with this protection.

Low-resistance pyrometers have often the leads of the same metals as the couples, so that the so-called "cold-junction" is at the terminals of the galvanometer, and the leads are then usually made long enough to permit the instrument being placed where the temperature can be maintained at about normal "room" temperatures. Variation in the "cold-junction" temperature from the calibration temperature produces more error in low-resistance than in high-resistance types.

When one wire of a couple is of iron, it should not be used for temperatures above 1300 to 1400°F., as this temperature is a "transition point" for this metal at which its physical and also its thermoelectric properties are changed.

Thermocouples made of platinum and a platinum-nickel alloy produce twice the voltage of a platinum-rhodium combination, but they should not be subjected to temperatures above 2000°F. If the couple is made of one wire of pure platinum and another of an alloy of platinum

and about 10 per cent rhodium, temperatures nearly as high as the melting point of platinum, or nearly 3500°F ., can be measured, although 3000°F . is considered the safe limit. This pyrometer with a platinum "couple" is known generally as a Le Chatelier type (Fig. 56).

Electric-resistance thermometers are based on the principle that the electrical resistance of some metals increases considerably as the temperature is raised. Platinum is usually selected because for a given temperature it has a remarkably constant resistance and it does not deteriorate at high temperatures. A resistance thermometer of the

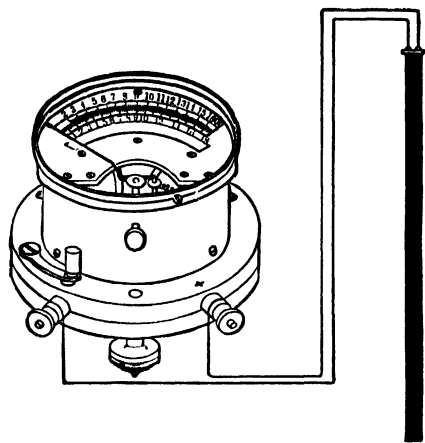


FIG. 56.—Le Chatelier pyrometer.

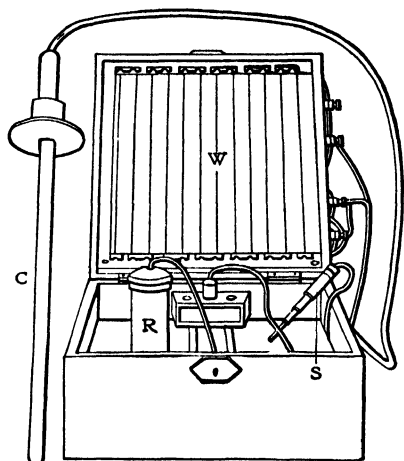


FIG. 57.—Resistance thermometer bridge.

simplest type is made of a coil of pure annealed platinum wire *W* wound upon a mica framework (Fig. 57) in "series" with a very small coil in a casing *C* intended to be exposed to the temperature to be measured. The variation of resistance is measured by the Wheatstone bridge method.¹ The current from one electric battery passes through the wire in *C*, and the current from another battery passes through the coil *W* in the cover of the box. When the two circuits are connected so that the electromotive forces of the two batteries are opposed, the resistance in the cover is adjusted by means of a connection on a stylus *S* so that there is no current passing through a telephone receiver *R* or a sensitive galvanometer placed at the junction of the two circuits. For making observations, this stylus is moved along the "scale" wire in the cover to a point where the humming noise due to the electric current ceases. The temperature can then be read on the graduated scale opposite the position of the stylus.

By means of a switchboard any number of "heating" elements can be connected to the same indicator box, which may be located at any distance from the source of heat.

¹ MARKS, "Mechanical Engineering Handbook," 3d ed., p. 2064.

Commercial instruments of this type are usually arranged so that the "bridge" indicates the temperature in degrees. Up to 1000°F. the error should be less than $\frac{1}{10}^{\circ}$ and at 2400°F. not more than $\frac{1}{2}^{\circ}$. A delicate galvanometer sensitive for small currents is required. A large current in the necessarily small wires would by its own heating change the resistance and impair sensitiveness. For many classes of work, particularly if there is rough usage, the platinum coil *C* must be protected by a porcelain or iron tube. This protection introduces a time lag, so that very delicate instruments are not protected by a casing. The junctions of the platinum wire of the thermometer with the wires going to the resistance-measuring device must be placed in the cooler part of the circuit, where the temperature should be the same as when the instrument was calibrated, or compensators may be used, as explained on page 63. Electric-resistance thermometers are readily *calibrated* at the temperatures of melting ice, by steam at varying pressures (from 212 to 350°F.), and by boiling sulphur (832.5°F.). Intermediate temperatures are computed. Metals like copper, tin, and zinc (page 53) when pure remain at a quite constant temperature for about a half hour when cooling slowly and passing from the liquid to the solid state.

Bimetal or mechanical pyrometers (Fig. 58) consist essentially of two rods made of metals having different rates of expansion connected

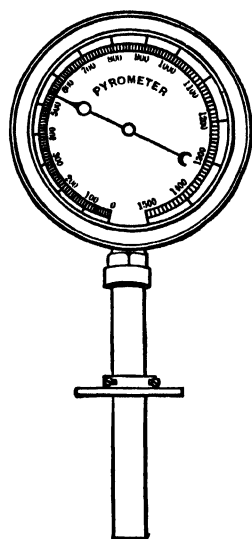


FIG. 58.—"Mechanical" pyrometer.

by gears and levers to rotate a pointer on a graduated dial. Generally the rods are made of iron and brass or of graphite and iron. Such instruments are generally very unreliable and should never be used for temperatures above 1000°F. There is always a tendency for the zero of the instrument to get higher with use. Beckert and Weinhold found that in a number of cases the zero changed from 200 to 400°F. in 2 months. In order to obtain readings corresponding to the graduations, the entire length of the tube enclosing the rods should be placed in the chamber the temperature of which is being measured. The Power Test Committee of the A.S.M.E. disapproves of their use.

Pneumatic pyrometers depend for their action on the variation of the flow of gases through orifices due to heating. Uehling's pneumatic pyrometer is shown diagrammatically in Fig. 59. As shown, flue gas is continuously drawn through two orifices *A* and *B* by a constant suction produced by an aspirator *D*. As long as the air has the same temperature in passing through *A* as it has in passing through *B*, there is no change in the partial vacuum

in the chamber between the two apertures; if, however, the air has a higher temperature when passing through *A* than when passing through *B*, the suction or vacuum in the chamber between the two orifices will increase in proportion to the difference in temperature between *A* and *B*, because the volume of air varies directly with the temperature.

In the application of this principle, orifice *A* is located in a nickel tube which is exposed to the heat to be measured, while orifice *B* is kept at a uniformly lower temperature. Filters are provided for keeping the orifices clean. The instrument can be made to record and indicate the temperature at a distance. In order to maintain a constant vacuum or suction at *B*, the steam pressure at the aspirator nozzle *D* must be maintained constant by a good reducing valve or other means.

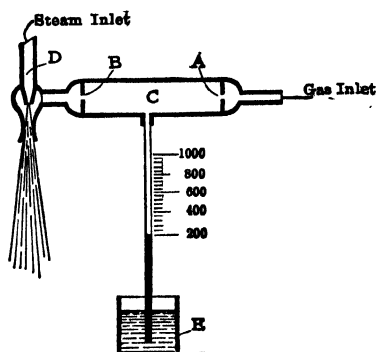


FIG. 59.—Pneumatic pyrometer.

Radiation Pyrometers.—For temperatures above 2500°F. radiation pyrometers similar to the one illustrated in Fig. 60 are most suitable. They can also be used in many places where it is almost impossible to

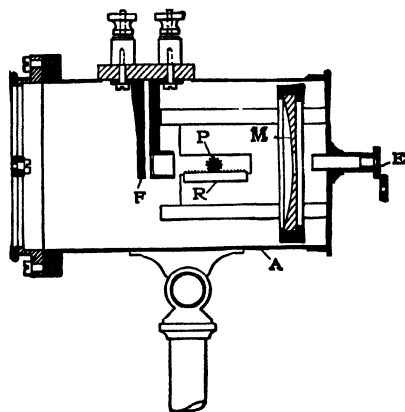


FIG. 60.—Telescope of Fery radiation pyrometer.

locate a pyrometer of any of the other types. The principle of operation is that the energy radiated by a so-called "black" body is proportional to the fourth power of its absolute temperature. The instrument illustrated consists of a cylindrical case set upon a tripod. This case contains a concave mirror and a lens (or lenses) which when properly adjusted and focused on a hot body concentrate the heat rays upon a small *thermo-electric couple*¹ inside the case. Copper wires connect this couple with a very sensitive portable galvanometer located where it can be read conveniently.

The most modern instruments of this kind are provided with scales indicating directly degrees of temperature. Figure 60 shows

¹ Some of these instruments have a metal coil made up of a pair of strips of metal of widely different coefficients of expansion which replaces the thermocouple. The principle of operation and the disadvantages are the same as those of bimetallic pyrometers (p. 66).

a section of the telescope used in connection with this pyrometer. The concave mirror *M* receives the heat rays and focuses them at *F*, where a small thermocouple is located. To assist in pointing the telescope, an eyepiece *E* is provided through which a reflected image of the hot body can be seen. The rack *R* and the pinion *P*, moved by a thumbscrew outside the case, serve for adjusting the focus of the mirror. In the center of the field of view, as seen in the eyepiece, the thermocouple is seen as a black spot, and this must be overlapped on all sides by the image of the hot body to obtain the correct temperature. It is interesting to observe that the distance of the telescope from the source of heat does not affect the reading of the instrument. When the telescope gets nearer the hot body, the mirror *M* receives, of course, more heat, but at the same time this greater amount of heat is distributed over a large image and the intensity of the heat remains the same.

Radiation pyrometers are calibrated in terms of the radiation from a so-called "black body," which is approximately realized by a uniformly heated enclosure. It is only for "black bodies," such as carbon and coal, that the temperature is exactly proportional to the fourth root of the heat energy. Readings obtained when measuring the temperature of a body not inside a closed chamber with hot walls will in some cases be very much lower than the true temperature. For a piece of heated coal the error is very small owing to lack of enclosure, while in molten copper or tin with a clean surface the temperature reading may be 100°F. too low. Conditions as regards enclosure are, however, satisfactory in most practical cases where the instrument is frequently used, such as taking the temperature of boiler furnaces, gas producers and retorts, annealing and hardening furnaces. Error due to the furnace door being open for an instant when the observation is to be made is practically negligible, especially as these instruments are actually calibrated under this condition. If excess of air in a furnace is likely to reduce the temperature while sighting, a large tube of cast iron or fire clay closed at the end toward the fire can be built into the furnace wall. By sighting through the open end upon the closed end, which should be at the furnace temperature, very satisfactory results are obtained.

Observations made with such pyrometers of incandescent bodies or gases do not give the true temperature. It is generally assumed, however, that they can be used to measure fairly accurately the temperature of heated chambers when focused upon the walls,¹ because of the reflection going on in all directions. In most cases the flame temperature can be taken as the same as that of the surrounding walls.

A relatively large area is usually required to sight radiation pyrometers. It is stated that the distance from the telescope to the hot body can be as much as thirty times the diameter of the hot body and the

¹ VON JÜPTNER, HANNS, "Heat Energy and Fuels," p. 76.

telescope can be taken as much nearer as desired without changing the reading of the instrument. Before taking observations the pointer of the galvanometer must be set at zero, the instrument receiving no heat rays during this adjustment. The readings of temperature made with such instruments are obviously the difference between the temperatures of the hot body and of the room.

Optical Pyrometers.—Another type of pyrometer, based in principle upon the measurement of the brightness of the hot body by comparison with a standard lamp, is shown in Fig. 61. In order to use this instrument, known as Wanner's, the incandescent (osmium filament) lamp

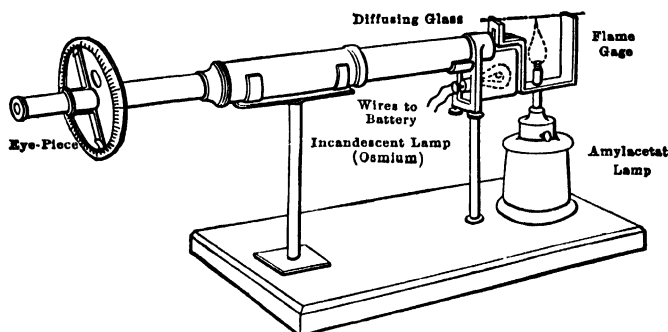


FIG. 61.—Wanner optical pyrometer in position for standardizing.

must first be standardized by comparison with an amylacetate oil lamp of constant candle power. Then after standardizing, it is necessary only to focus the instrument upon the hot body to be measured and the temperature is read directly on the graduated scale at the eyepiece.

Temperature readings from optical pyrometers are actual and are not differences depending on the temperature of the room.

Furnace temperatures may be determined approximately from the values corresponding to the color of the fire. All temperatures are in degrees Fahrenheit.

Red—just visible.....	900	Orange.....	2000
Dull red.....	1250	White.....	2350
Cherry red.....	1600	Dazzling white.....	2700

Radiation and optical pyrometers are invaluable for determining the temperatures of the various parts of a furnace, of the walls of the setting of a steam boiler, of various portions of a bed of coal, etc. It is sometimes stated that an optical pyrometer is a means for measuring temperatures of objects "miles away."

Calorimetric Pyrometers.—If the specific heat and weight of a body are known, its temperature can be obtained by observing the rise in temperature of a known quantity of water into which the body is thrown.

More in detail, the method consists in the determination of temperature by putting a ball of metal or other refractory material into the medium of which the temperature is to be measured. When the ball has become heated uniformly throughout its mass to the temperature of the medium, it is transferred quickly to a cup heavily jacketed with non-conducting material in which there is a known weight of water at a known temperature. Copper, wrought iron, and fire clay are suitable materials. Specific heats of these materials at about 500°F. are, respectively, 0.097, 0.110, and 0.180. Since metals are attacked by furnace gases, they should be protected when used in this way in a crucible of refractory material.

This method, although not recommended for special tests, is often very serviceable in places or at times when accurate pyrometers are not available. On account of the "personal" error liable to enter, such determinations should be repeated several times to **check** the results. Calculations required are as follows:¹

Let w_1 = weight of the ball, in pounds.

w_2 = weight of the cup (only the "inner" vessel),² in pounds.

w_3 = weight of the water in the cup, in pounds.

t_1 = initial temperature of water, in degrees Fahrenheit.

t_2 = final temperature of the water, in degrees Fahrenheit.

t_0 = temperature of the heated ball, in degrees Fahrenheit.

s_1 = specific heat of the ball.

s_2 = specific heat of the cup.

Then

$$w_1 s_1 (t_0 - t_2) = (w_2 s_2 + w_3) (t_2 - t_1),$$

$$t_0 = \frac{(w_2 s_2 + w_3) (t_2 - t_1)}{w_1 s_1} + t_2. \quad (3)$$

Pyrometer Cones.—For many purposes when a pyrometer cannot be well placed, fusible pyrometer cones are used. Such cones are made of ceramic material mixed so as to give a definitely known melting point for each one. The melting points range from 1120 to 3650°F. by irregularly unequal steps, each having a standard number. These cones are carefully graded, so that, if one has had some experience with them, temperatures can be **estimated** to about the nearest 15 to 20°F. Four of these cones are shown in Fig. 62.

When a series of cones is placed in a furnace, the one having the lowest melting point begins to turn over first. The temperature corresponding

¹ A more complete description of calorimetric pyrometers and the precautions to be observed for accuracy will be found in *Trans. A.S.M.E.*, Vol. 6, p. 712.

² It would be more accurate, of course, to use in the calculation the water equivalent of the whole vessel, as is done in coal calorimetry (see p. 222). Units given are in pounds and degrees Fahrenheit, but other units, provided they are corresponding, can be used in the equation given.

to the cone number is reached when the tip of the cone has bent over and just touches the mounting "pat" on which it is standing. Hence the highest temperature reached when the cones shown in the illustration were used was about halfway between that corresponding to each of the two middle cones. According to the numbers on the cones as shown in the figure the temperature, as given by the following table, was between 1526 and 1580°F. The greatest disadvantage with this system is that there is no way of observing a decrease in the temperature, or, in other words, only the maximum temperature is indicated.

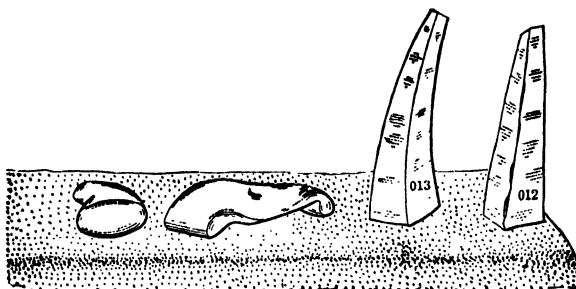


FIG. 62.—Pyrometer cones after use.

For regulating the temperature in kilns used for making refractory products such as brick, fire clay, or high-temperature mortar, carefully graded pyrometer cones are used for fairly accurate temperature determinations. In order to determine temperature with these cones, a series laid out to include the approximate temperature to be measured is set in the place where the test temperature is desired. When a set of cones is heated, they melt one after another, either completely or partly, and the required temperature is that corresponding to the number in Table XII, of the cone of which the tip just touches the mounting pat on a level with the base of the cone.

Pyrometer cones do not fuse sharply but soften over a temperature range having a beginning and an *end point*. The end point is reached just before a cone is fused. In this country the pyrometer cones made for Professor Orton by the Standard Pyrometer Cone Company, Columbus, Ohio, are generally used. Table XII gives the end points¹ for Orton's standard pyrometric cones.

Precautions in the Use of Pyrometer Cones.—For maximum accuracy the Orton cones should be set on mounting pats of refractory material of a composition that will not affect the end points of the cones. A suitable

¹ End-point temperatures are accurate only when the cones are heated under certain standard conditions usually stated as a rate of 270°F. per hour. The temperatures of end points given in Table XII are mean values, for which a maximum difference from the mean of not more than 10°F. may be expected up to 2500°F., and of not more than 10 to 18°F. from that temperature up to the highest value given in the table.

TABLE XII.—END POINTS OF ORTON CONES

Cone No.	End point, degrees Fahrenheit	Cone No.	End point, degrees Fahrenheit	Cone No.	End point, degrees Fahrenheit
022	1121	02	2057	19	2768
021	1139	01	2093	20	2786
020	1202	1	2120	23*	2876
019	1220	2	2129	26	2903
018	1328	3	2138	27	2921
017	1418	4	2174	28	2939
016	1463	5	2201	29	2984
015	1481	6	2246	30	3002
014	1526	7	2282	31	3056
013	1580	8	2300	32	3092
012	1607	9	2345	33	3173
011	1661	10	2381	34	3200
010	1643†	11	2417	35	3245
09	1706	12	2435	36	3290
08	1742	13	2462	37	3308
07	1814	14	2552	38	3335
06	1859	15	2615	39	3389
05	1904	16	2669	40	3425
04	1940	17	2687	41	3578
03	2039	18	2714	42	3659

* Cones 21, 22, 24, and 25 are not available.

† There is a negative increment between cones 011 and 010.

material for making these pats is ground alundum mixed with Norton No. 518 cement, using in the mixing as little water as possible. The pyrometer cones should be mounted with the base embedded $\frac{1}{8}$ inch in the pat and the face of one side inclined at an angle of about 80 degrees to the horizontal. The pat is usually circular about $2\frac{1}{4}$ inches in diameter and at least $\frac{1}{2}$ inch thick. The cones may be attached to the pat with the same kind of cement that was used for making the pat.

Figure 63 represents a Beckmann **differential thermometer**, which is unique in that its range can be shifted by displacing part of the mercury into a reservoir at the top. This is used in calorimeters where a thermometer of short range and one that is capable of being read to a small fraction of a degree is required. By the use of a low-power telescope it is possible to read these thermometers to 0.001°C . Thermometers of this type have a large time lag, and it is necessary to make allowance for this in some cases. The range of the thermometer may be adjusted as required. The length of the scale is generally equivalent to 5°C . and is arbitrarily numbered from 0 to 5.

Armored Thermometers.—Two types of mercury thermometers protected by heavy metal cases are illustrated by Figs. 64a and 64b. These are sometimes called armored industrial thermometers. It will be

observed that a threaded thermometer well is a part of the casing. The one shown in Fig. 64b has graduations for reading both **temperatures and pressures**. A thermometer of this type is particularly useful in pipes carrying hot boiler-feed water. When the temperature is above 212°F., the thermometer will indicate that the water is being heated at a pressure higher than atmospheric. For water heated in closed vessels or pipes

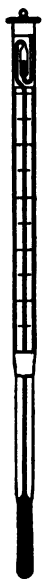


FIG. 63.—Beckmann differential thermometer.

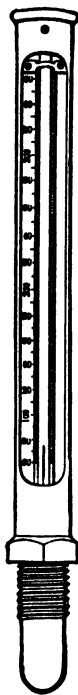


FIG. 64a.—Armored thermometer.

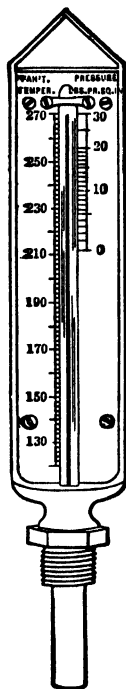


FIG. 64b.—Combined thermometer and pressure gage for boiler feed water.

there is for every temperature a corresponding pressure as given in tables of the properties of saturated steam.¹

Precautions for Accuracy in Temperature Measurements.—Before inserting a thermometer or pyrometer in the place where it is to be used for temperature determination, the following precautions as given by the Power Test Committee of the A.S.M.E. should be observed: (1) For accurate results with a thermometer, it should be entirely immersed and the surface of the well through which the thermometer extends should not be abnormally heated or cooled. The use of excessively long emergent stems of thermometers should be avoided. In steam lines the thermometer should not be near heavy pipe flanges that are not well

¹ Tables of the properties of saturated steam are given in Table I of the Appendix.

insulated. (2) Emergent-stem corrections cannot be accurately calculated and safely applied to temperature readings from armored or industrial (page 72) thermometers. The emergent-stem corrections should be obtained by the comparison method (page 51) with bare

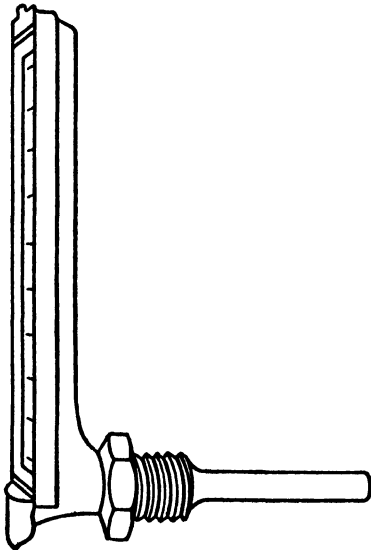


FIG. 64c.—Angle-stem armored thermometer.

thermometers. (3) If the top of the liquid thread in the capillary tube of a thermometer has separated itself from the main thread which extends into the bulb, a correction can be subtracted by estimating the length of the separating space in degrees of the thermometer calibration; and this amount must be subtracted from all the thermometer readings taken at the top of the mercury thread in order to obtain the correct reading for temperature. (4) Electric lights that will not be turned off between readings should not be placed near the stem of a thermometer, if the heat from the lamp is an appreciable amount. This precaution is especially necessary when measuring relatively low temperatures, as, for example, near or below the freezing point of water.

(5) Avoid parallax in reading thermometers.

Alcohol Thermometers.—For the measurement of temperatures much below zero Fahrenheit, thermometers filled with mercury are not satisfactory, and alcohol or “spirits of wine” is used. These liquids, on the other hand, are not suited, on account of their high vapor tensions, for high temperatures. Alcohol of average purity freezes at about 50°F. below zero.

Conversion of Temperatures and Heat Units.—Temperatures in Centigrade degrees are converted into Fahrenheit by multiplying by $\frac{9}{5}$ and adding 32. Kilogram-calories multiplied by 3.968 give the equivalent British thermal units (B.t.u.), and kilogram-calories per kilogram $\times 1.8$ (the *temperature scale factor*) give British thermal units per pound. A “small” or gram-calorie is one-thousandth as large as a kilogram-calorie.

One horsepower-hour is equivalent to 2,545 British thermal units (B.t.u.) per hour or 42.42 British thermal units per minute. Similarly, one kilowatt-hour is equivalent to 3,413 British thermal units per hour.

CHAPTER III

DETERMINATION OF THE MOISTURE IN STEAM

Unless the steam used in the power plant is superheated, it is said to be either dry or wet, depending on whether or not it contains water in suspension. The general types of **steam calorimeters**, used to determine the amount of moisture in the steam, may be classified under four heads:

1. Throttling or superheating calorimeters.
2. Separating calorimeters.
3. Electric calorimeters.
4. Condensing (barrel) calorimeters.

Throttling or Superheating Calorimeters.—The type of steam calorimeter used most in engineering practice operates by passing a sample of the steam through a small orifice, in which it is superheated by throttling. A calorimeter of this kind can be made of pipe fittings as illustrated in Fig. 65. It consists of an orifice O , discharging into a chamber C , into which a thermometer T is inserted, and a mercury manometer is usually attached to the cock V_2 , for observing the pressure in the calorimeter.

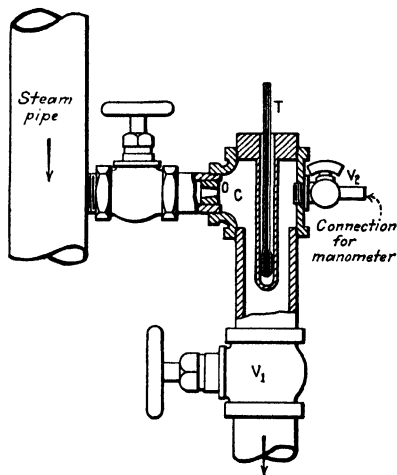


FIG. 65.—Simple throttling calorimeter.

It is most important that all parts of calorimeters of this type, as well as the connections leading to the main steam pipe, should be very **thoroughly lagged** by a covering of good insulating material. One of the best materials for this use is hair felt, and it is particularly well suited for covering the more or less temporary pipe fittings, valves, and nipples through which steam is brought to the calorimeter. The hair-felt **lagging should be at least 2 inches thick**. Many throttling calorimeters have been declared useless by engineers and put into the scrap heap merely because the small pipes leading to the calorimeters were not properly lagged, so that there was too much radiation, producing, of course, condensation, so that the calorimeter did not get a true sample. It is obvious that, if the entering steam contains too much moisture, the drying action due to the throttling in the orifice may not be

sufficient to superheat. It may be stated, in general, that, unless there is about 5 to 10°F. of superheat in the calorimeter or, in other words, unless the temperature on the low-pressure side of the orifice is at least about 5 to 10°F. higher than that corresponding to the pressure in the calorimeter, there may be some doubt as to the accuracy of results.¹ The working limits of throttling calorimeters vary with the initial pressure of the steam. For 35 pounds per square inch absolute pressure the calorimeter ceases to superheat when the percentage of moisture exceeds about 2 per cent; for 150 pounds absolute pressure, when the moisture exceeds about 5 per cent; and for 250 pounds absolute pressure, when it is in excess of about 6 per cent. For any given pressure in the main the exact limit varies slightly, however, with the pressure in the calorimeter.

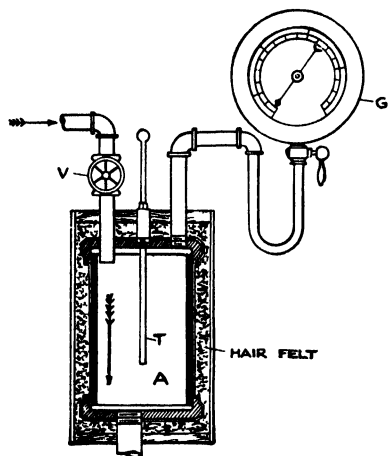


Fig. 66.—Peabody throttling calorimeter.

The throttling calorimeter (Fig. 66) devised by Prof. C. H. Peabody, when properly used, gives very accurate results.²

In connection with a report on the standardizing of engine tests, the A.S.M.E. published the following instructions regarding the method to be used for obtaining a fair sample of steam from the main pipes. It is recommended in this report that the calorimeter shall be connected with as short intermediate piping as possible with a so-called **steam calorimeter sampling nozzle**.

Steam-sampling Nozzles (A.S.M.E. Recommendations, 1930).—With most calorimeters of the types described, it is quite possible, generally speaking, to determine the quality of wet steam with the degree of accuracy usually required. Calorimeters, however, do not necessarily indicate the quality of steam flowing in a pipe, even if they do show

¹ The same general statement may be made as regards determinations of superheat in engine and turbine tests. Experience has shown that tests made with from 0 to 10°F. superheat are not reliable, and that the steam consumption in many cases is not consistent when compared with results obtained with wet or more highly superheated steam. The errors mentioned, when they occur, are probably due to the fact that in steam, indicating less than 10°F. superheat, water in the liquid state may be taken up in "slugs" and carried along without being entirely evaporated.

² Professor E. H. Lockwood of Yale University states that he has made tests with the simple throttling calorimeter (Fig. 65) parallel with A.S.M.E. type (p. 84) and has found differences in superheat of 15°F. or more.

Professor J. J. Eames of the Massachusetts Institute of Technology states that in his opinion the Peabody calorimeter is more reliable than most other types.

the moisture in the sample which they receive. The determining factor in accurate quality determinations is quite likely to hinge on the ability to obtain a representative sample; and this is especially true when the moisture content is comparatively high. The sampling nozzle recommended by the A.S.M.E. is shown in Fig. 67.

The installation of all types of calorimeters with the exception of the external superheating (Fig. 75a, page 92) requires but a single connection with the steam main. Such a connection is made directly to the sampling nozzle which has been inserted in the main. The external superheating calorimeter requires the use of two connections with the main, one to the sampling nozzle and the other directly to the main. The direct connection is for the purpose of obtaining the steam which is to be externally superheated.

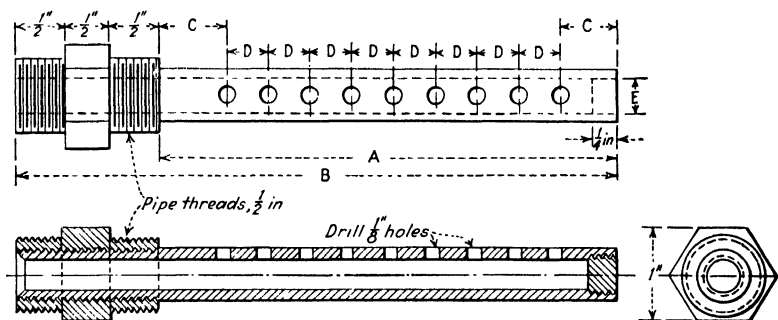


FIG. 67.—A.S.M.E. standard steam-sampling nozzle.

If the sample of steam is to be obtained from a line 4 inches or greater in diameter, it is necessary only to drill and tap a hole to fit the bushing into which the sampling nozzle is to be screwed.

If the sample is to be obtained from a line less than 4 inches in diameter, it is usually necessary to insert a T pipe fitting similar to Fig. 40, page 45, if one is not already available.

The location of this nozzle is a matter of prime importance and, in general, it should be placed where the flow is in parallel stream lines. If at all possible, a vertical pipe should be selected. It has been shown by experiment that a vertical pipe in which the steam descends is the most satisfactory. In a horizontal pipe, for example, the moisture content is likely to be concentrated toward the bottom. The ratio of the densities of the moisture and steam in the pipe may be as great as several hundred. It can be seen, therefore, that it is highly important to locate the sampling nozzle at a point where the steam and water are well mixed. The descending steam main meets this condition better than any other.

The sampling nozzle for pressures above atmospheric should be made of $\frac{1}{4}$ - or $\frac{3}{8}$ -inch pipe, preferably the smaller. The part of the nozzle to be placed inside the main should be drilled in a straight line with $\frac{1}{8}$ -inch holes

spaced as given in Table XIII. One end of the sampling nozzle (Fig. 67) should be threaded to fit the bushing to which the calorimeter or intermediate piping is to be attached. The other end, or the end to be placed inside the main, should be plugged as shown. The following table gives the numerical values of the dimensions lettered in Fig. 67 for the common sizes of "extra heavy" steam piping. In the case of "standard full-weight" piping the internal diameter is greater than for "extra heavy" piping, but the same nozzle may be used in either case. Nozzles for the sampling of low-pressure steam have not been standardized.¹

TABLE XIII.—DIMENSIONS OF STEAM SAMPLING NOZZLES FOR PRESSURES ABOVE ATMOSPHERIC

(For Use in "Extra Heavy" Wrought-iron and Steel Pipes)

All dimensions are given in inches

Nominal size of line	Internal diameter	Number of holes in nozzle	Diameter holes	A	B	C	Center distance of holes, D	Pipe size of nozzle, E
2	1.939	3	$\frac{1}{8}$	$1\frac{3}{4}$	$3\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{4}$
2½	2.323	3	$\frac{1}{8}$	2	$3\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$
3	2.9	5	$\frac{1}{8}$	$2\frac{3}{4}$	$4\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{4}$
3½	3.364	5	$\frac{1}{8}$	3	$4\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$
4	3.826	6	$\frac{1}{8}$	$3\frac{5}{8}$	$5\frac{1}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{1}{4}$ or $\frac{3}{8}$
4½	4.290	6	$\frac{1}{8}$	4	$5\frac{1}{2}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{1}{4}$ or $\frac{3}{8}$
5	4.813	6	$\frac{1}{8}$	$4\frac{5}{8}$	$6\frac{1}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{1}{4}$ or $\frac{3}{8}$
6	5.761	6	$\frac{1}{8}$	$5\frac{1}{2}$	7	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{4}$ or $\frac{3}{8}$
8	7.625	6	$\frac{1}{8}$	$7\frac{1}{2}$	9	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$ or $\frac{3}{8}$
10	9.75	6	$\frac{1}{8}$	$9\frac{1}{2}$	11	$\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$
12	11.75	7	$\frac{1}{8}$	$11\frac{1}{2}$	13	$\frac{1}{2}$	$1\frac{3}{4}$	$\frac{3}{8}$
14	13	7	$\frac{1}{8}$	$12\frac{1}{2}$	14	$\frac{5}{8}$	$1\frac{7}{8}$	$\frac{3}{8}$
16	14.75	8	$\frac{1}{8}$	$14\frac{1}{4}$	$15\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{7}{8}$	$\frac{3}{8}$
18	16.5	8	$\frac{1}{8}$	16	$17\frac{1}{2}$	$1\frac{7}{16}$	$1\frac{7}{8}$	$\frac{3}{8}$
20	18.5	9	$\frac{1}{8}$	18	$19\frac{1}{2}$	1	2	$\frac{3}{8}$

The nozzle should be installed in the line in such a position that the holes will directly face the steam flow. It is not always possible, however, to insert the sampling nozzle in the most desirable location. The ideal location is a pipe in which the steam flow is vertically downward and as far removed as possible from any valve, elbow, or other disturbing element ahead of the nozzle. The second choice is a pipe in which the steam ascends vertically with other conditions the same as already described. In this case, the results will be very erratic at low steam velocities, particularly if there is an orifice or restriction in the main below the sampling tube. Under such conditions the condensation on the sides of the steam line will run back against the flow of steam and will be

¹ See p. 90 for an example of a low-pressure nozzle.

"caught" by the higher velocity steam at the restricted portion and will be thrown upward into the calorimeter sampling nozzle. An installation with the sampling nozzle in such a position has to be used with great care. The pipe in which the steam ascends will usually show more moisture, with the same nozzle and steam, than when the steam descends, but it is not possible to recommend a correction factor.

Pipe bends and horizontal pipes should at all times be avoided, if an accurate determination of the quality of the steam is to be obtained. If, however, the horizontal pipe cannot be avoided, the sampling nozzle should be placed in a horizontal position close to a valve (as shown in Fig. 67a), so that the disturbing influence will serve to mix thoroughly the water and steam.

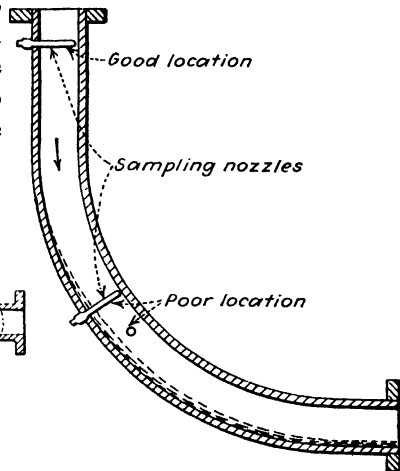
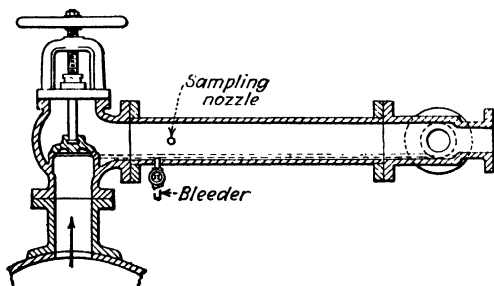


Fig. 67a.—Sampling nozzle in horizontal pipe. Fig. 67b.—Sampling nozzles in pipe bend.

In the case of a bend, provided no other location is possible, the sampling nozzle should be placed in the straight part at the entering end of the bend and in the plane of the bend, as indicated in Fig. 67b.

Materials for Nozzles.—Brass is a suitable material for nozzles to be used in steam up to 500°F. Nozzles made from plain carbon steels (sulphur, maximum 0.05 per cent) may be used for temperatures greater than 500°F.

Figure 67c shows locations of sampling nipples from which erroneous results will be obtained.

Case 1. Horizontal Pipe.—Water flows at the bottom of the pipe. If there are perforations in the nipple near the bottom of the pipe, water piles against the nipple, flows into the calorimeter, and gives false readings.

Case 2.—If the nipple is located too near the junction of two horizontal runs of pipe as at *a*, condensation from the vertical pipe which collects at this point will be thrown against the nipple by the velocity of the steam, resulting in false readings. The nipple should be located far enough above the junction to be removed from water kept in motion by the steam velocity, as at *b*.

Case 3.—Condensation in the pipe bend will be held by the velocity of the steam as shown. When the velocity is diminished during firing intervals, for example, water flows back against the nipple *a* and a false reading is obtained. A true reading will be obtained at *b*, provided condensation is not blown over upon the nipple.

Case 4.—Where a nonreturn valve is placed in front of a pipe bend, condensation will collect on the steam-line side and water will be swept by the steam velocity against the nipple and false readings will result.

Never close and usually do not attempt to adjust the discharge valve V_1 of the steam calorimeter in Fig. 65 without first closing the gage cock V_2 . The same precaution applies to closing the cock under the gage G of the calorimeter shown in Fig. 66 under the same circumstances. Unless this precaution is taken, the pressure may be suddenly increased in the calorimeter chamber so that if a manometer is used the mercury will be blown out of it, and if, on the other hand, a low-pressure steam gage is used, it may be ruined by exposing it to a pressure much beyond its scale.

Usually it is a safe rule to begin to take observations of temperature in calorimeters after the thermometers have indicated a maximum value and have again receded slightly from this maximum.

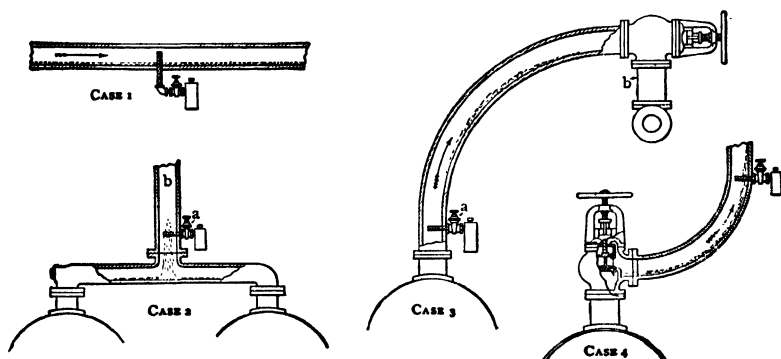


FIG. 67c.—Wrong places for steam calorimeters.

Calculation of Steam Quality by Throttling Calorimeters.—The quality or relative dryness of wet steam is easily calculated by the following method. Using the symbols,

p_1 = steam pressure in main, in pounds per square inch absolute

p_2 = steam pressure in calorimeter, in pounds per square inch absolute.

t_c = temperature in calorimeter, in degrees Fahrenheit.

h_{g1} and h_{f1} = heat (enthalpy)¹ of evaporation of steam and heat (enthalpy) of liquid (sensible heat) corresponding to pressure p_1 , in B.t.u. per pound.

h_{g2} and t_2 = total heat (enthalpy) (B.t.u. per pound) and temperature (degrees Fahrenheit) of steam corresponding to pressure p_2 .

c_p = specific heat of superheated steam. Assume 0.46 for approximately atmospheric pressures in calorimeters.²

¹ See p. 361.

² In the A. S. M. E. Test Code for Steam Generating Units (Cf. Item 1,112, p. 318), the corresponding value as given is 0.47. Average values for the specific heat of superheated steam for any pressures and temperatures are given on p. 368.

x_1 = initial quality of steam (a decimal).

$100(1 - x_1)$ = initial moisture in steam, in per cent.

Total heat in a pound of wet steam flowing into the orifice is

$$x_1 h_{f01} + h_{f1},$$

and after expansion, assuming all the moisture is evaporated, one finds the total heat of the same weight of superheated steam to be

$$h_{02} + c_p(t_c - t_2).$$

Then, assuming no heat losses, we have

$$x_1 h_{f01} + h_{f1} = h_{02} + c_p(t_c - t_2),$$

or

$$x_1 = \frac{h_{02} + c_p(t_c - t_2) - h_{f1}}{h_{f01}}. \quad (4)$$

Charts for Moisture Determinations.--A small section of the total-heat-entropy chart as provided in modern steam tables is shown in Fig.

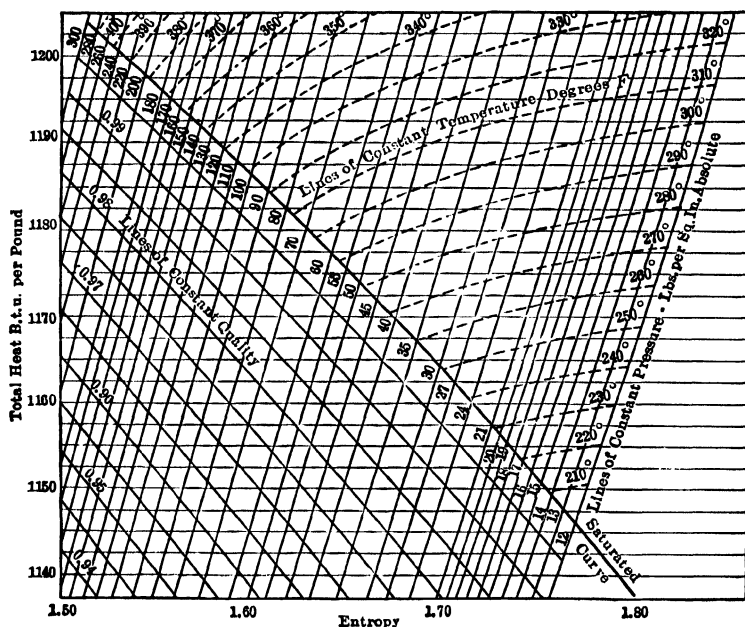


FIG. 68.—Chart for determining quality of steam with any throttling calorimeter.

68. It is arranged particularly for determinations of the quality of steam with a throttling calorimeter without using the equations above. Horizontal lines in the chart are those of constant total heat of the steam, and represent the process in a throttling calorimeter. To illustrate the application of the chart, let the initial pressure of steam be 165 pounds per square inch absolute and the reading of the thermometer on the

low-pressure side of the calorimeter be 270°F. The pressure in the calorimeter is 15.2 pounds per square inch absolute. To find the quality x , start at the intersection of the temperature line for 270° with the 15.2 pounds pressure line and go across the chart horizontally to the 165 pounds line, then the "lines of constant quality" indicate that the quality of the steam is 0.979.

When a U-tube manometer is used to determine the pressure in a well-lagged calorimeter of the type illustrated in Fig. 65, this pressure can be obtained very accurately, and an excellent means is provided for **calibrating the thermometer** in the calorimeter just as it is to be used. The calibration would be made, of course, by the method of comparing with the temperature corresponding to known pressures explained on page 55. In order to avoid having superheated steam in the calorimeter for this calibration, the felt or similar material usually needed for covering the valves and nipples between the main steam pipe and the calorimeter should be kept saturated with cold water.

The Barrus Throttling Calorimeter.—An important variation from the type of throttling calorimeter shown in Fig. 65 has been introduced

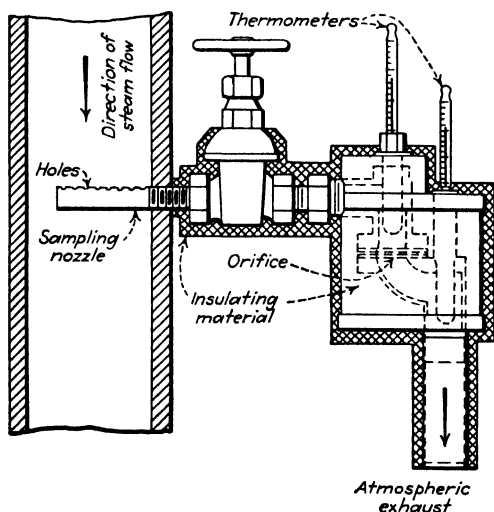


FIG. 69.—Barrus throttling calorimeter with standard A.S.M.E. sampling nozzle.

quite widely by George H. Barrus. In this apparatus the **temperature** of the steam admitted to the calorimeter is observed instead of the pressure and a very free exhaust is provided, so that the pressure in the calorimeter is atmospheric. This arrangement simplifies very much the observations to be taken, as the quality of the steam x_1 can be calculated by Eq. (4) by observing only the two temperatures t_1 and t_c , taken respectively on the high- and low-pressure sides of the orifice in the calorimeter. This calorimeter is

illustrated in Fig. 69. The two thermometers required are shown in the figure. Arrows indicate the path of the steam.¹

The orifice in such calorimeters is usually made about $\frac{3}{8}$ inch in diameter; and for this size of orifice the weight of steam² discharged per

¹ *Trans. A.S.M.E.*, Vol. 11, p. 790.

² Formulas for calculating the exact weight of steam discharged from a nozzle are given on pp. 195 and 197. In boiler tests, corrections should be made for the steam

hour at 175 pounds per square inch absolute pressure is about 60 pounds. It is important that the orifice should always be kept clean, because if it becomes obstructed there will be a reduced quantity of steam passing through the instrument, making the error due to radiation relatively more important.

In order to free the orifice from dirt or other obstructions the connecting pipe or calorimeter nipple to be used for attaching the calorimeter

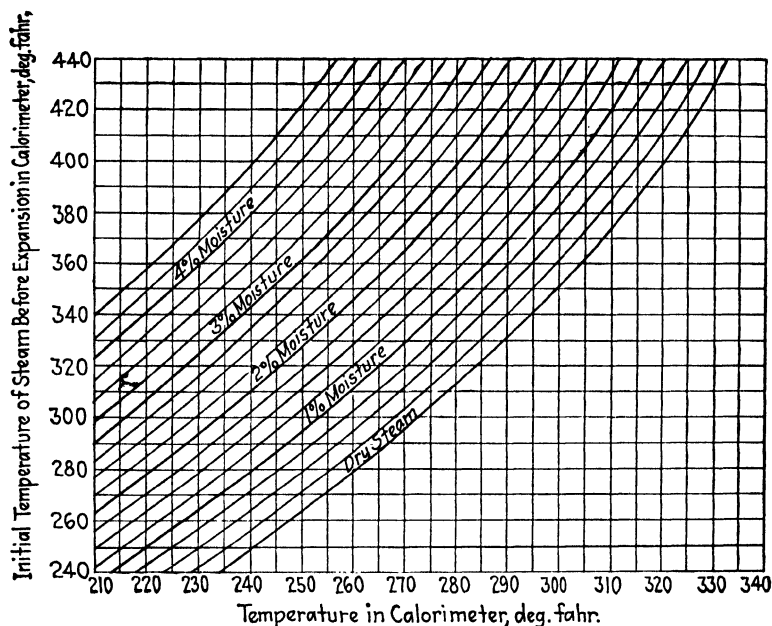


FIG. 69a.—Chart for determining quality of steam from temperatures. (Atmospheric pressure in calorimeter.)

to the main steam pipe should be blown out thoroughly with steam before the calorimeter is put in place. The connecting pipe and valve should be covered with hair felting not less than 1 inch thick. It is desirable also that there should be no leak at any point about the apparatus, either in the stuffing box of the supply valve, the pipe joints, or the union.

With the help of such a diagram the Barrus calorimeter is particularly well suited for use in power plants, where the quality of the steam is entered regularly on the log sheets. The percentage of moisture is obtained immediately from two observations without any calculations.

discharged from the steam calorimeters. The Power Test Committee of the A.S.M.E. suggest the use of Napier's formula (p. 195), believing it to be sufficiently accurate for this kind of work.

degrees representing 1 per cent of moisture. To determine this "normal" reading corresponding to dry steam, the instrument should be attached to a horizontal steam pipe in such a way that the sampling nozzle projects upward to near the top of the pipe, there being no perforations, and the steam enters through the open top of the nozzle. The test should be made when the steam in the pipe is in a quiescent state, and the steam pressure is maintained constantly at the point observed on the main trial. If the steam pressure falls during the time the observations are being made, the test should be continued long enough to obtain the effect of an equivalent rise of pressure.

To find the constant for 1 per cent of moisture, divide the latent heat of the steam supplied to the calorimeter at the observed pressure or temperature by the specific heat of superheated steam at about atmospheric pressure and then divide the quotient by 100. Finally ascertain the percentage of moisture by dividing the number of degrees of cooling by the "constant" as above noted.

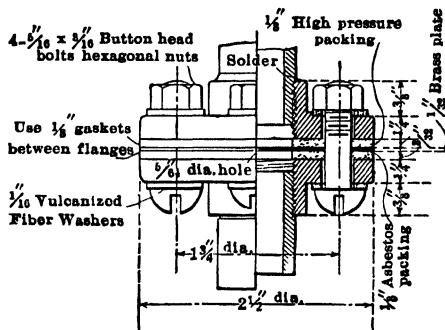


FIG. 70a.—Detail of orifice for Fig. 70.

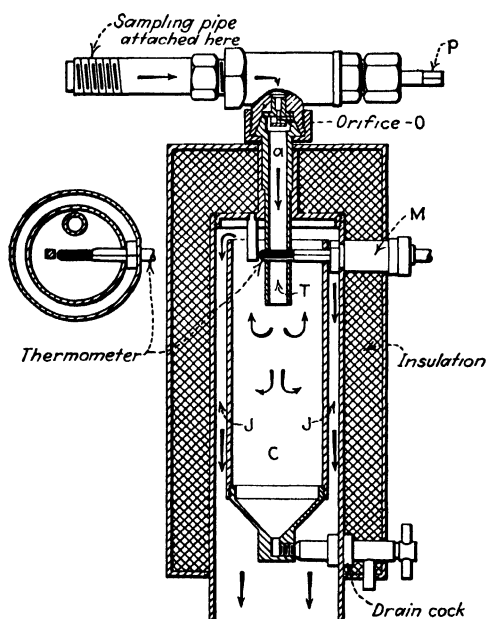


FIG. 71.—Ellison U-path calorimeter.

The Ellison U-path steam calorimeter is shown in Fig. 71. This calorimeter is similar to the Barrus type (page 82), as the orifice *O* reduces the steam pressure to approximately atmospheric. Below the orifice, the steam impinges against a plane surface which divides it into four atomized streams. These four atomized streams, however, meet again after being deflected and are mixed below the orifice plug. The throttled steam flows in the steam chamber *C* of the calorimeter in a path resembling the letter U (down one side and up the other) and discharges at the top, where its direction of flow is again turned so that it flows downward through the

steam jacket *J, J* and out at the bottom to the atmosphere. In order to

reduce the heat radiation losses, the jacket *J, J* is insulated by filling a 1-inch space on the sides and a $1\frac{1}{2}$ -inch space at the top with lamp black.

In this calorimeter, the position of the thermometer is unusual, as it is inserted from the side directly into the path of the steam, through the tube *T*, and is held in place there by means of a suitably designed packing cup *M*.

Figure 72 shows a section of the Crosby two-pressure steam calorimeter.

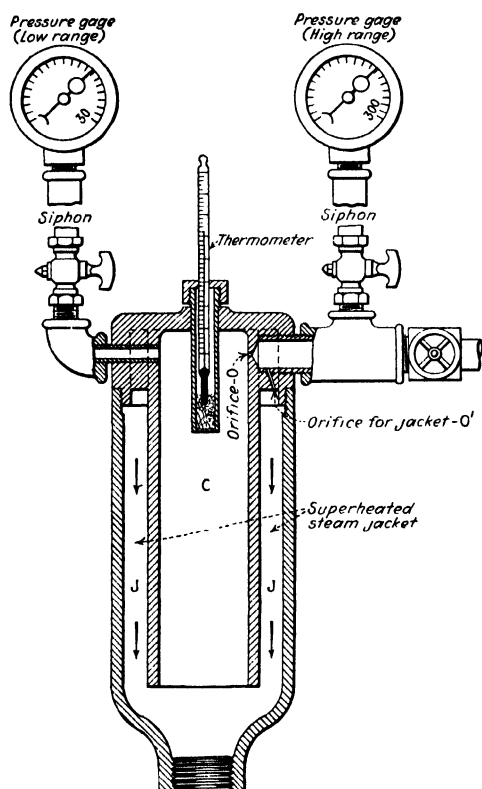


FIG. 72.—Two-pressure throttling calorimeter.

rimeter is made of bronze and highly polished on the outside and has an unusually good steam jacket, the heat radiation loss may be regarded as negligible.

Separating Calorimeters.—It was explained on page 76 that throttling calorimeters cannot be used for the determination of the quality of steam when for comparatively low pressures the moisture is in excess of 2 per cent, and when for average boiler pressures in modern engineering practice it exceeds 5 per cent. For higher percentages of moisture than these low limits, separating calorimeters are most generally used. In these instruments the water is removed from the sample of steam by mechani-

eter. This is an unusual design, as it is provided with *two* orifices, *O* and *O'*. One of the orifices (*O*) supplies steam to the chamber *C* in which the temperature is measured, while the other orifice (*O'*) supplies steam to the downflow jacket *J, J* surrounding the inner chamber *C*. The steam in the jacket is thus at the same temperature as the steam within the chamber *C*, where the temperature is taken. The pressures in the chamber *C* and the jacket *J, J* are likewise equal, so that there is little possibility of the jacket temperature's influencing the temperature in the steam chamber. The flow is downward, thereby preventing any condensation from collecting, either in the chamber *C* or in the jacket.

In the A.S.M.E. Power Test Code, it is stated that, because the body of this calorimeter

cal separation, just as it is done in the ordinary steam separator installed in the steam mains of a power plant. There is provided, of course, a device for determining, while the calorimeter is in operation, usually by means of a calibrated gage glass, the amount of moisture collected. This mechanical separation depends for its action on changing very abruptly the direction of flow and reducing the velocity of the wet steam. Then since the moisture (water) is nearly three hundred times as heavy as steam at the usual pressures delivered to an engine, the moisture will be deposited because of its greater inertia.

One of the simplest forms of separating calorimeters, made of pipe fittings, is shown in Fig. 73. Steam enters at *A*, passes down through the vertical pipe *P*, plugged at the lower end, from which it escapes through a large number of $\frac{1}{8}$ -inch holes indicated in the figure. In passing through these holes, the direction of flow is changed very abruptly, since the steam must go upward to be discharged at *D*. Moisture is deposited at the bottom of the vessel *V*, and its volume or weight can be determined from the height of the water in the gage glass *G* if the vessel has been calibrated. Steam discharged from *D* must be condensed and weighed in a pail or barrel containing cold water. The percentage of moisture is then found by dividing the weight of water collected in the vessel *V* by the sum of the weight of steam condensed and the weight of water collected in *V*. This sum is, of course, the weight of the **wet steam**.

Radiation Loss.—As in all calorimetry work, in order to obtain accurate results there should be a covering of hair felt 1 inch thick over **all parts** of the apparatus, and even then the radiation loss is sometimes large enough to make corrections necessary.¹ This correction is determined by operating two calorimeters which are exactly alike in construction and in the amount of felt covering, in series, and so arranged that the second takes the discharge of the first. If it is known that the discharge from the first calorimeter is perfectly dry steam,² then the moisture collected in

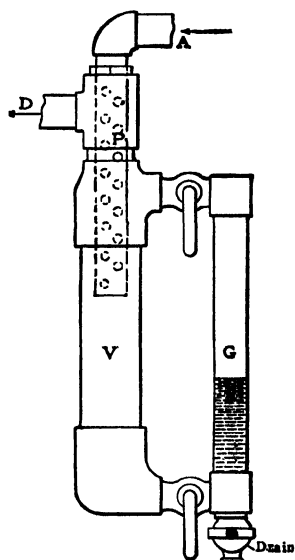


FIG. 73.—A simple separating steam calorimeter made of pipe fittings.

¹ A steam calorimeter like Fig. 65 in which no lagging is used in its construction must be very carefully protected all over by at least a 2-inch thickness of the best kind of heat-insulating material. A 1-inch thickness is usually sufficient for the calorimeters in Figs. 66 and 69.

² A small throttling calorimeter can be attached to the discharge from the first separating calorimeter to determine by a separate test whether or not the steam discharged is dry, but not superheated.

the second calorimeter is the condensation due to its own radiating surface, which should be the same as for the first. Calculations for corrected moisture determinations are made then by subtracting from the moisture collected in the first instrument the amount condensed in the second. When, of course, the radiation loss has been once determined,

it is not necessary to operate the second calorimeter.

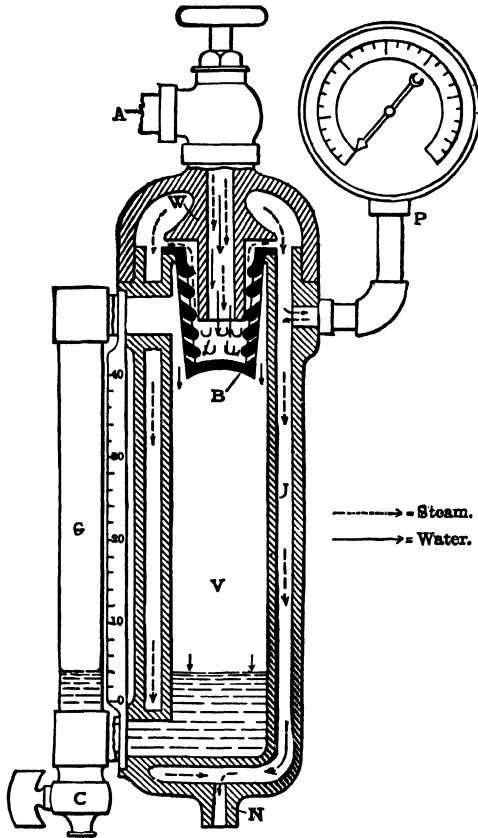


FIG. 74.—Separating steam calorimeter with steam jacket.

Figure 74 illustrates a form of separating calorimeter in which the improvement over the one shown in Fig. 73 is in the addition of a steam-jacketing space receiving live steam at the same temperature as the sample. Steam is supplied through a pipe *A*, discharging into a cup *B*. Here the direction of the flow is changed through nearly 180 degrees, causing the moisture to be thrown outward through the meshes of the cup into the vessel *V*. The dry steam passes upward through the spaces between the webs *W* into the top of the outside jacketing chamber *J* and is finally discharged from the bottom of this steam jacket through the nozzle *N*. This nozzle is considerably smaller than any other section through which the steam flows, so that there is no appreciable difference be-

tween the pressures in the calorimeter proper and the jacket. The scale opposite the gage glass *G* is graduated to show in **hundredths of a pound**, at the temperature corresponding to steam at ordinary working pressures, the variation of the level of the accumulating water. A steam pressure gage *P* indicates the pressure in the jacket *J*, and, since the flow of steam through the nozzle *N* is roughly proportional to the pressure (see page 195), another scale in addition to the one reading pressures is provided at the outer edge of the dial; this scale indicates pounds of dry steam per hour. A petcock *C* is used for draining the water from

the instrument, and, by weighing the water collected corresponding to a given difference in the level in the gage *G*, the scale opposite it can be readily calibrated. Too much reliance should not be placed on the readings for the flow of steam as indicated by the gage *P*, unless it is frequently calibrated. Usually it is very little trouble to connect a tube to the nozzle *N* and condense the steam discharged in a large pail nearly filled with water. When a test for quality is to be made by this method, the pail nearly filled with cold water is carefully weighed, and then, at the moment when the level of the water in the water gage *G* has been observed, the tube attached to the nozzle *N* is immediately placed under the surface of the water in the pail. The test should be stopped before the water gets so hot that some weight is lost by "steaming." The gage *P* is generally calibrated to read pounds of steam flowing in 10 minutes. For the best accuracy, it is desirable to use a pail with a tightly fitting cover into which a hole just the size of the tube has been cut.

Combined Separating and Throttling Calorimeters.—Calorimeters as already described remove practically all the moisture in steam (page 87) when the pressure is not lower than 25 pounds per square inch by gage. For lower pressures, particularly around atmospheric, recent experiments show that the efficiency of such calorimeters is in some cases not more than 80 per cent.¹ For this reason in the best current practice for determinations of moisture in low-pressure steam a throttling calorimeter is attached to the discharge of the separating calorimeter. Then if the separating calorimeter has been carefully calibrated for radiation loss and the steam escaping from the separating calorimeter is tested again in a throttling instrument, it is possible to make correct determinations for the percentage of moisture in the steam of almost any degree of wetness. An apparatus of this kind which is reported to have done excellent service in tests of very large low-pressure steam turbines, operating with the exhaust from reciprocating steam engines in New York City, is shown in Fig. 75. The most notable feature of this apparatus is the sampling tube. It was found that for this *low-pressure steam* the ordinary sampling tube of perforated pipe as shown in Fig. 67 did not give a reliable sample. It was also found necessary that the sample should be taken from the main without changing its direction or velocity until it is actually inside the sampling pipe. If the direction of flow of wet steam is suddenly changed when entering the sampling nozzle, the entrained moisture, because of its greater specific gravity on the one hand and the very slight skin friction between it and the surrounding dry steam on the other, will cause it to continue in its path in a straight line, so that there is a tendency for only dry steam to enter the nozzle.

¹ *Trans. A.S.M.E.*, Vol. 32, p. 1132. The efficiency of the calorimeter is the ratio of the percentage of moisture taken out by the separating calorimeter to the total percentage of moisture.

Also if the velocity of the steam in the sampler is greater than that in the main, there is a tendency for the dry steam to "accelerate" into the nozzle, leaving the moisture behind. It has been stated that this action has not been observed in tests of steam at high pressures, because of (1) smaller differences between the specific gravity of high-pressure steam and water, (2) greater skin friction, (3) the highly divided state of the moisture.¹

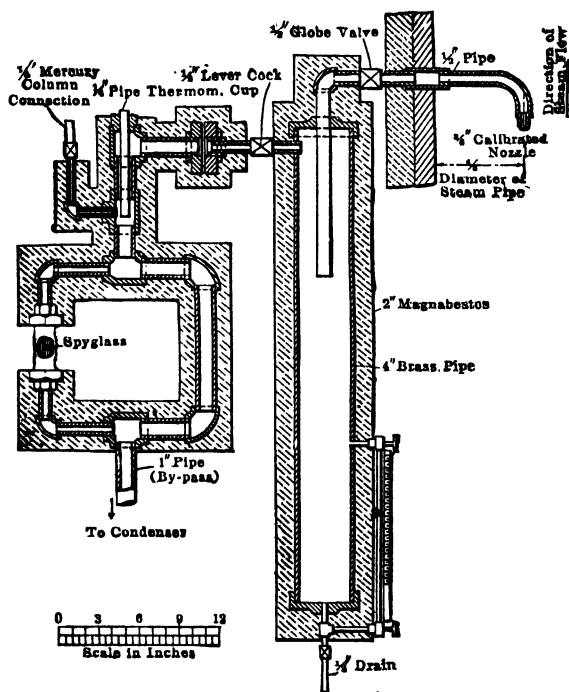


FIG. 75.—Combined separating and throttling steam calorimeter.

As the throttling calorimeter is ordinarily used, it would have very little capacity when used with steam pressures only a little above atmospheric; but by making it discharge into a receiver in which a vacuum of about 28 inches of mercury is maintained, the throttling portion of the calorimeter will evaporate from 2 to 3 per cent of moisture.

The apparatus shown in Fig. 75 consists of the $\frac{3}{8}$ -inch brass nozzle on the sampling tube, which is bent to point in the direction opposite to the flow of the steam. The lip of this nozzle is filed to a knife edge to avoid disturbing the current of steam around the mouth of the sampler by eddies and impact against a thick lip. This sampling tube is set up so that it extends into the main steam pipe one-sixth of the diameter of the pipe, where it has been observed to give practically a true average

¹ STOTT, H. G., and R. J. S. PIGOTT, *Trans. A.S.M.E.*, Vol. 32, p. 77.

sample. When in operation, the valve at the sampling tube is opened wide and the flow is regulated by means of the lever cock between the separating and the throttling calorimeters. The necessary throttling action ordinarily produced by an orifice is produced by this cock. A vacuum is maintained in the throttling portion from which the discharge is carried to a small cooling receiver in which the steam is condensed. From this receiver it flows to a "volumetric" measuring tank, the top of which is tightly closed and connected by $\frac{1}{4}$ -inch pipes to the main condenser. A spyglass shown at the left in the figure is useful for proving that the calorimeter is working properly. It often happens that, when the superheat in the calorimeter is less than from 5 to 8°F., there is some moisture passing through, and the spyglass will invariably show it. As the spyglass is most conveniently made of $\frac{3}{4}$ -inch gage glass, its area is not large enough to carry all the steam, and a by-pass connection is arranged as shown. The large size of the parts is necessary on account of the very large specific volume of the low-pressure steam.

All parts of the apparatus are carefully covered with magnesia-asbestos covering 2 inches thick. For the normal rate of flow for the instrument, the radiation can be made less than 0.1 per cent.

Calculation of Percentage Moisture for Combination Separating and Throttling Calorimeter.—Quality of steam x_1 is calculated for a combination calorimeter as follows:

Let w_1 = weight of moisture collected in the separating calorimeter in a given time, in pounds.

w_2 = weight of dry steam condensed after passing through the throttling calorimeter, in pounds.

x_2 = quality of steam discharged from separating portion as determined by the throttling calorimeter.

Then without sensible error,

$$1 - x_1 = \frac{w_1}{w_1 + w_2} + (1 - x_2)$$

and in terms of "quality" (always a decimal), we have, **more accurately**,¹

$$x_1 = x_2 \times \frac{w_2}{w_1 + w_2} \quad (5)$$

Superheating Calorimeter (External Type).—A steam calorimeter arranged for superheating the steam by means of an external source of heat is shown in Fig. 75a. It consists of a supply pipe S_1 , which is to be suitably connected to the sampling nozzle (page 77), which furnishes the

¹ If the radiation test shows it is large enough to be appreciable, and if R is weight of condensation due to radiation in pounds in a given time corresponding to that for the other units, then

$$x_1 = x_2 \times \frac{w_2 + R}{w_1 + w_2}$$

steam to be tested, and another steam pipe S_2 , also connected to the main steam supply but not through the sampling nozzle, which supplies steam to the superheater H . The

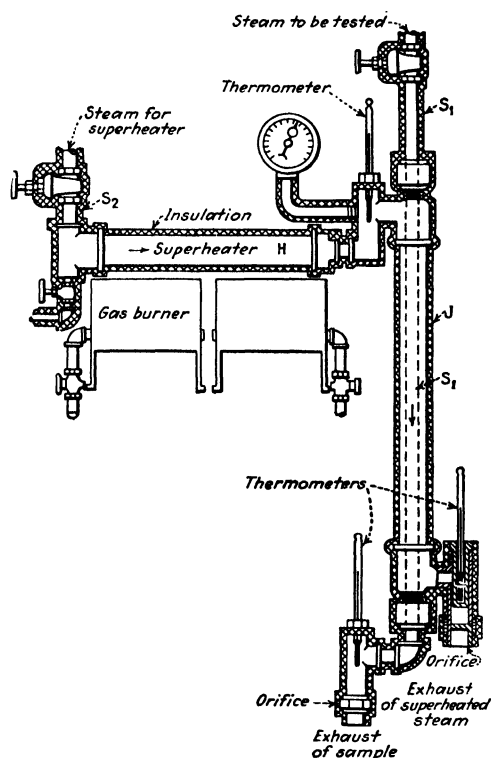


FIG. 75a.—External superheating calorimeter.

the two branches for steam supply (S_1 and S_2) should be made to carry *equal quantities of steam*. Then if radiation is neglected, the heat loss by the superheated steam must equal that gained by the test sample pound per pound. For calculating the results of tests of steam quality x made with this calorimeter, the following equation is used:

$$x = \left[1 - \frac{W_1(h_{g3} - h_{g4}) - (h_{g2} - h_{g1})}{h_{f01}} \right] \times 100$$

where W = weight of saturated steam flowing through calorimeter in stated time, in pounds.

W_1 = weight of superheated steam flowing through jacket in stated time, in pounds.

¹ The amount of heat received from the external source for superheating the steam, as from a gas burner, does not enter into the calculation of the quality of the steam.

t_1 = temperature of saturated steam corresponding to absolute pressure in the line, in degrees Fahrenheit.

h_{g1} = total heat (enthalpy¹) of saturated steam corresponding to absolute pressure in the line, in B.t.u. per pound.

h_{g2} = total heat (enthalpy) of steam sample leaving calorimeter, in B.t.u. per pound.

h_{g3} = total heat (enthalpy) of superheated steam entering jacket, in B.t.u. per pound.

h_{g4} = total heat (enthalpy) of superheated steam leaving jacket, in B.t.u. per pound.

R = radiation heat loss, in B.t.u. in the stated time.

x = quality of steam, in per cent.

h_{fg1} = heat (enthalpy) of evaporation at the temperature t_1 , in B.t.u. per pound.

If W equals W_1 , and the correction for radiation (R) is included, the final equation becomes

$$x = \left[1 - \frac{(h_{g3} - h_{g4}) - (h_{g2} - h_{g1}) - R}{h_{fg1}} \right] \times 100.$$

In the use of these calorimeter equations the time element does not enter, provided W equals W_1 .

Electric Steam Calorimeters.—

For use with very wet steam the Thomas electric calorimeter (Fig. 76) has been designed. It consists essentially of a cylindrical vessel B containing a series of resistance coils of German silver wire for heating steam by means of the electric current passing through them. These coils are connected to the electric terminals or binding posts shown in the figure and are supported in a soapstone cylinder in which there are a large number of $\frac{1}{4}$ -inch holes through which the coils pass up and down.

Steam enters at the bottom of the vessel at A and, passing upward through the heated coils, the moisture contained in it is evaporated. The steam then passes up through a perforated casing filled with copper gauze and escapes through the pipe discharging at the side at C . A part

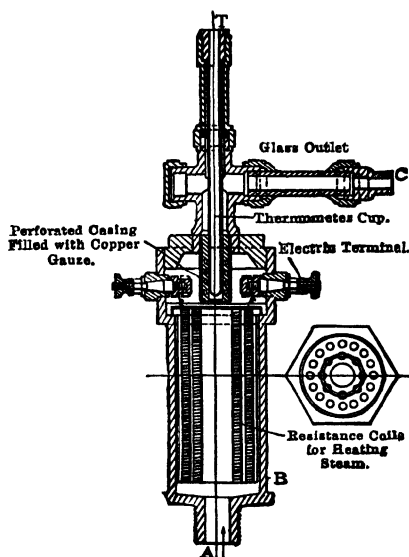


FIG. 76.—Electrically-heated steam calorimeter.

¹ The notation used here is the same as that in the Steam Tables (Table I in the Appendix).

of this latter pipe is made of a glass tube for observing the condition of the steam. A thermometer is inserted at T for observing the temperature after this reheating.

In the operation of the instrument a steady flow of steam must first be secured; then the electric current should be turned on to dry the steam. When it is dry or superheated, the fog will disappear from the glass observation tube. The first step is to superheat the steam to some temperature T , requiring an electrical input of E_t watts. Then decrease the current until the steam is just dry, requiring an input of E_0 watts. Then $E_t - E_0 = E'$, which represents the watts required to superheat the steam t° to the temperature T . By applying a constant K , determined by experiments for a series of pressures and degrees of superheat, the following equation is obtained:

$$h = \frac{KE_0}{E},$$

where h is the heat (B.t.u.) required to dry a pound of steam. A series of curves giving values of K are supplied with each instrument. If h_{fg} is the heat (enthalpy) of evaporation per pound of the dry saturated steam corresponding to the pressure, then the quality

$$x = \frac{h_{fg} - h}{h_{fg}}.$$

These curves make the use of this calorimeter practicable but the accuracy is not so great (by from 1.5 to 2.5 per cent) as that attainable by condensing and weighing the steam.

The advantage that this calorimeter possesses is its wide range. Its disadvantages are: (1) difficult to manipulate; (2) expensive, especially if wattmeter is included in the cost; (3) possible to obtain superheat readings without being able to get the steam absolutely dry.

Although this apparatus is used for steam of high quality as well as low, it has not been generally used, probably because throttling calorimeters are preferred on account of greater simplicity in both construction and operation, and because very often a source of electric current is not conveniently available where tests are to be made. No data are available comparing its efficiency with that of the combined separating and throttling calorimeters described in the preceding paragraphs, but for **accurate** tests the latter are generally preferred by engineers.

Errors in this type of instrument are likely to be due to a variable weight of steam discharging in a unit of time; that is, the weight discharged will be less for superheated than for dry saturated steam. If, however, the steam discharged is condensed and weighed, the error from this source can be eliminated. It is practically impossible to make enough curves of values of K for all the variables.

In the A.S.M.E. Power Test Code, it is stated that this electric-conductivity method of determining the moisture content of steam is based on the premise that the electric conductivity of pure water is zero, but actually the conductivity of a condensed-steam sample varies directly with the total amount of solids present in the sample of steam being tested.

Condensing or Barrel Calorimeters.—There is still another kind of steam calorimeter, known as the Hoadley barrel type, deserving some attention. It is one of the oldest forms

of apparatus for making determinations of the quality of steam. In the classification made at the beginning of this chapter it belongs in the group of **condensing** calorimeters. A typical apparatus as shown in Fig. 77 consists of a weighing barrel *B*, made of three concentric vessels of galvanized iron with the two annular spaces between the inner and outer vessels filled with pressed sheet cork or hair felt to reduce radiation to a minimum. It is usually

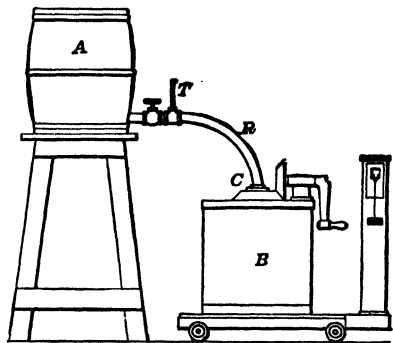


FIG. 77.—“Barrel” steam calorimeter.

arranged so that, when the inner vessel has been nearly filled with water from the barrel *A*, a quantity of the steam to be tested can be passed into it. The steam is admitted into the barrel *B* in the most common forms by disconnecting the water hose *R* at *C* and making a temporary connection from the steam pipe out of which a sample is to be taken to a vertical pipe in the barrel of sufficient length to extend nearly to the bottom of the inner vessel. The pipe may be plugged at the lower end and sufficient area for the escape of steam is then secured by drilling into the pipe a number of $\frac{1}{8}$ -inch holes for some distance from the lower end. This arrangement will make it easier to secure an equal rise in temperature in the different parts of the barrel. A float is usually provided to show the depth of water in the barrel, and a suitable stirring device or agitator consisting of paddles attached to a vertical shaft is also needed. This agitator, when revolved, stirs up the water and brings it to a constant temperature.

Briefly, the method to be pursued in the operation of the barrel calorimeter may be outlined as follows: First, fill the barrel with cold water until the float shows that the water level is about 6 inches from the top. Then stir well, observe the temperature accurately, and weigh carefully on a platform scales. The steam pipe should then be connected up to discharge into the water after first allowing the steam to blow off into the air, not only to remove the condensation in the piping but also to heat it to as nearly as possible the temperature of the steam. When

the temperature has risen to about 120°F., the steam should be shut off and another weighing made to determine the amount of steam added. While the weighing is being done, the water should be stirred vigorously and the highest temperature observed. For all the weighings the piping must be in exactly the same position as regards the connection on the barrel and all the pressure in the pipes must be relieved. When the piping between the calorimeter and the steam supply is connected by pipe fitters' unions, these should be disconnected to insure the best accuracy. When, however, the connection is made by means of a very flexible rubber hose, the weight can be obtained accurately enough without disconnecting the piping if the precaution is taken to relieve the pressure in the piping by opening a petcock located in the steam pipe near the point where it enters the barrel.

If, just before making the test for the quality of the steam, the calorimeter is filled with water, heated with steam or otherwise to about 150°F. and again carefully drained, the barrel will be near its average temperature during the test, and no correction need probably be made for the heat absorbed by the calorimeter. In most cases it is preferable, however, to determine accurately the heat absorbed by the calorimeter and then make the proper corrections; but unless the work is done very carefully, it is valueless. This correction is usually made by calculating the **water equivalent** or the capacity of the calorimeter to absorb heat measured by the similar capacity of water. The water equivalent is to be added to the weight of water in the calorimeter. If the following symbols are used:

w' = weight of water in calorimeter, in pounds.

w'' = weight of water added, in pounds.

t' = temperature of water in calorimeter, in degrees Fahrenheit.

t'' = temperature of water added (higher than t'), in degrees Fahrenheit.

t''' = temperature of mixture, in degrees Fahrenheit.

k = water equivalent, in pounds.

Then

$$(w' + k)(t''' - t') = w''(t'' - t'''),$$

$$k = \frac{w''(t'' - t''')}{t''' - t'} - w'. \quad (6)$$

The temperature of the water added should be taken just as it **enters** the calorimeter and as near to it as possible.

The quality of the steam (x_0) to be determined is calculated as follows:

w_1 = weight of water in calorimeter, in pounds.

w_2 = weight of steam added, in pounds.

k = water equivalent of calorimeter, in pounds.

t_1 = initial temperature of water in calorimeter, in degrees Fahrenheit.

- t_2 = final temperature of water, in degrees Fahrenheit.
 p_0 = pressure of steam, in pounds per square inch.
 h_{f0} = heat of evaporation of steam (B.t.u.) corresponding to p_0
 h_{f0} and h_{f2} = heat of liquid (sensible heat) of steam (B.t.u.) corresponding to p_0 and t_2 .

Then equating the heat lost by the steam to the heat gained by the water.¹

$$(x_0 h_{f0} + h_{f0} - h_{f2})w_2 = (w_1 + k)(t_2 - t_1)$$

$$x_0 = \frac{(w_1 + k)(t_2 - t_1)}{w_2 h_{f0}} + \frac{h_{f2} - h_{f0}}{h_{f0}}. \quad (7)$$

The accuracy of this instrument depends mainly on the care with which the various temperatures and the weight of the condensed steam are obtained. Usually it is very difficult to obtain accurately the temperature of the mixtures of water and steam. It is not unusual for determinations of *moisture* with such a calorimeter to vary for the same quality of steam and with expert handling as much as 5 per cent. In the case, therefore, of steam with 10 per cent moisture, the determination of *quality* might be in error as much as $\frac{1}{2}$ per cent.

In the A.S.M.E. Power Test Code, it is stated that the barrel type of steam calorimeter is very inaccurate, and its use is not permitted in any of the "code" tests.

Comparison of Steam Calorimeters.—The types of throttling calorimeters are "ideally suited for accurate determinations" of the quality of wet steam.² This type of calorimeter, even in the hands of inexperienced observers, or with gages and thermometers that are not intended for high accuracy, is yet an instrument with which comparatively high accuracy can be obtained, if it is assumed, of course, that the sample of steam entering the calorimeter is of the average quality. In fact, in all cases of the use of steam calorimeters, one must realize that the largest errors are likely to occur from incorrect sampling of the steam. In comparison with other types, especially the electric calorimeter, the superheating calorimeter (external type), and the universal calorimeter, the throttling type is very simple, light in weight, low in cost, and has small radiation losses.

The separating, the universal (throttling-separating), and the superheating (including electric) have a very large range for quality determinations, while the range of a throttling type is relatively small. It may also be added that throttling calorimeters, as commonly made

¹ Equation (7) can be made a little simpler for calculations without appreciable error by writing for $h_{f2} - h_{f0}$ the difference between the corresponding temperatures ($t_2 - t_0$).

² A.S.M.E. Power Test Code, Instruments and Apparatus, Part XI, p. 6.

commercially, are defective, in that the orifice through which the high-pressure steam expands is too small, and, consequently, there is a small error in quality determinations, tending to make the results too low. In order to overcome this disadvantage, it is suggested in the A.S.M.E. Power Test Code (Instruments and Apparatus, Part XI, page 6) that the throttling orifice, expressed as *area*, should not be less than 0.0004 of the *cubic contents* of the expansion chamber.

Because of greater accuracy, comparatively small size, portability, and general indestructibility, throttling calorimeters are generally preferred by engineers. In the order of their relative accuracy, steam calorimeters are usually classed as follows: (1) throttling; (2) combined separating and throttling; (3) separating; (4) superheating; (5) electric; (6) barrel.

Calorimeter Calibrations.—For a laboratory calibration exercise three calorimeters of different types are connected by means of exactly the same kind of fittings and valves to the same steam main or receiver. A water jacket or a device like that shown in Fig. 48 should be provided to vary the quality of the steam. Tests should be made simultaneously and for the same length of time on the three instruments.

CHAPTER IV

MEASUREMENT OF AREAS

Two methods are in general use for obtaining the area of irregular figures like indicator diagrams: (1) **by measuring ordinates** and (2) **by means of a planimeter**. Variations of the method of ordinates are known as trapezoidal, Durand's, and Simpson's. To find the area by any of these methods, divide the figure into an even number of strips by parallel lines. The accuracy is increased as the number of strips is made larger. The notation used in the formulas is illustrated in Fig. 78, where y_0 is the length of the first ordinate, y_1 of the second, etc., n is the

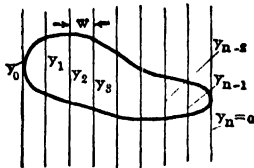


FIG. 78.—Diagram of ordinate methods.

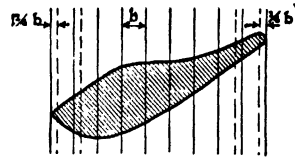


FIG. 79.—Granberg's "line pattern."

number of strips, w is the common width of the strips, and A is the area of the figure. Then the following approximate formulas may be stated:

I. By trapezoidal rule,

$$A = w(\frac{1}{2}y_0 + y_1 + y_2 + \dots + y_{n-1} + \frac{1}{2}y_n).$$

II. By Durand's rule,

$$A = w(0.4y_0 + 1.1y_1 + y_2 + y_3 + \dots + y_{n-2} + 1.1y_{n-1} + 0.4y_n).$$

III. By Simpson's rule,

$$A = \frac{1}{3}w(y_0 + 4y_1 + 2y_2 + 4y_3 + \dots + 2y_{n-2} + 4y_{n-1} + y_n).$$

A very convenient method of measuring areas by the use of a "line pattern" has been devised by Granberg.¹ A sheet of tracing cloth or thin celluloid is prepared with parallel lines on it, equally spaced, and with dotted lines (Fig. 79) as shown, located at each end of the figure at one-fourth the distance between the unbroken lines first drawn. This "line pattern" is then laid upon the area to be measured so that the ends of the area fall on the "solid" parallel lines at opposite ends. The

¹ GRANBERG, "Technische Messungen," p. 48.

sum of the lengths of the "solid" lines included by the outline of the area added to one-half the sum of the lengths of the two dotted lines included at the ends when multiplied by the distance between the parallel "solid" lines b gives the required area.

The various lengths required for both Granberg's and the trapezoidal rules can be conveniently added by laying them off with a dividers one after the other along a straight line and finally measuring the total length of the line.

Areas are also frequently calculated by the **method of mean ordinates**, as given on page 152, for finding the mean effective pressure in engine cylinders.

Planimeters.—The most accurate and generally approved method of obtaining the area of irregular figures is by means of integrating instruments called planimeters. Instruments of this kind may differ in many details, yet all of them are based in theory on the original Amsler polar planimeter.

Polar Planimeters.—One of the simplest forms of the polar type of planimeters is shown in Fig. 80. It consists essentially of two arms PO

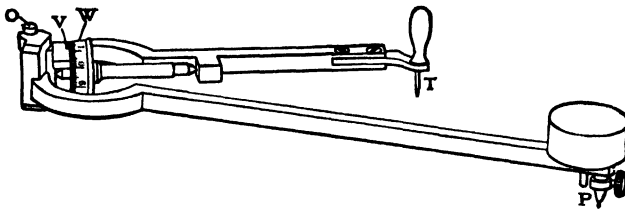


FIG. 80.—Amsler polar planimeter.

and TO pivoted together at O . When in use, the point P is not to be moved and is held in place by means of a pin point upon which a small weight rests. There is a **tracing point** at T intended to be moved around the border of the area to be measured. Attached to the arm TO is a small graduated wheel W carried on a short axis which must be placed accurately parallel to TO . Any movement of the arm TO except in the direction of its axis will, of course, move the wheel W on the paper or other surface on which it is placed in such a way that the amount of its movement gives a record indicating the area measured. A vernier V placed opposite the graduations on the wheel assists in reading the instrument accurately. The arm TO is usually made of such a length that the movement of the tracing point T around an area of 1 square inch (for English units) will move the wheel one-tenth of its circumference. Graduations of the vernier indicate usually one one-thousandth of a revolution of the wheel, or in English units one one-hundredth of a square inch.

When the tracing point T is moved around an area in a clockwise direction, the wheel will roll in the direction of its graduation, and the

area is found by subtracting the final reading from the initial. Amsler planimeters are often constructed with the arm TO adjustable in length, so that it can be set to indicate areas in various units, as, for example, square inches, square feet, square centimeters, etc.

The vernier V has 10 graduations, and the total length of these 10 divisions is one-tenth less than the length of those on the wheel, so that it represents, counted from zero, so many hundredths of a square inch. To explain the method of using the vernier, Fig. 81 has been inserted, showing the wheel W and the vernier V in a drawing of larger scale than in Fig. 80. Readings of the graduations on the wheel W are always

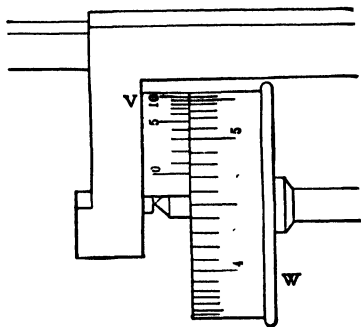


FIG. 81.—Typical vernier for a planimeter.

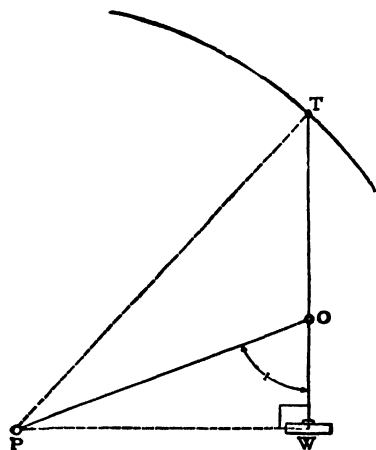


FIG. 82.—Position of the arms of a polar planimeter to draw the "zero" circle.

taken opposite the zero mark on the vernier, so that the reading indicated in Fig. 81 without the help of the vernier would be a little more than 4.7. The graduation on the vernier which is exactly coincident with a graduation on the roller wheel is the third from zero and indicates three hundredths. The complete reading is therefore 4.73 as determined by the vernier.

Theory of Polar Planimeters.—As this instrument is constructed, neither of the points T and W can pass over the arm PO (Fig. 82). If the arms PO and TO are clamped so that the plane of the graduated wheel W intersects the point P ,¹ that is, when the angle TWP is a right angle, and then the arms thus clamped are revolved around this point, the wheel will be continually slipping without any rolling motion in the direction of its axis, and consequently it will not revolve. When, however, the arms are not clamped and if the construction of the instrument will permit the tracing point T to be moved out so far that the axis of W will lie in the line PO , then an arc described by the movement of T will

¹ As regards the theory, it is immaterial whether W is between O and T or on TO extended. Some planimeters are made one way and some the other.

produce only a **rolling** motion of the wheel. Obviously, with the arms in any position intermediate between that of the clamped right angle and the one with W in line with PO , the wheel will partly slip and partly roll, the amount of slipping and rolling depending on the size of the angle between the arms. It follows, then, that when circumscribing a closed figure the radial components cause only slipping of the wheel and need not be considered, while the circumferential components produce a resultant rolling which must be taken into consideration.

The path described by the tracing point T when the arms are clamped, as indicated in Fig. 82, is called the **zero circle** of the planimeter.

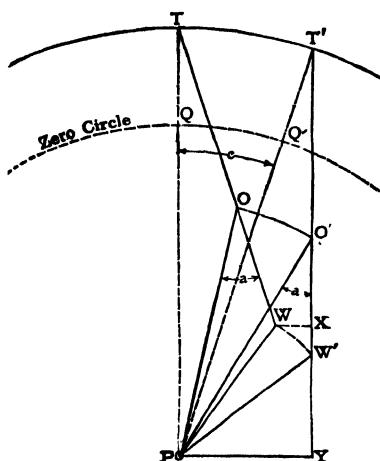


FIG. 83.—Theoretical diagram for a polar planimeter.

If the tracing point is moved in any path outside the zero circle in a clockwise direction, a positive record will be indicated on the graduated wheel; while if it is moved in a path in the same direction as before but inside the zero circle, there will be a negative record.

According to the theory of polar planimeters, they are designed so that the rolling of the wheel for a given circumferential motion of the tracing point T is proportional to the area included between the path of T , the radial lines from P (Fig. 83) to the initial and final points of the path taken by T , and the arc of the zero circle included between these radial lines. In other words, the

area referred to is $QTT'Q'$ in Fig. 83. In the discussion of this theory, the circumferential motion of the tracing point T around the point P , with the angle WOP (marked a) remaining always at a constant value, is to be taken up first.¹ Now let us suppose the tracing point is moved from T to T' in the figure through a very small angle TPT' (marked e), keeping, however, the angle a constant; then the graduated wheel W will move through the arc WW' , partly rolling and partly slipping. The component of this motion producing **rolling** will be perpendicular to the axis of the wheel; or, in other words, this component will be perpendicular to OT in all its positions, and without appreciable error for small values it may be represented in this figure by the line WX , making WXW' a right-angled triangle of infinitesimal proportions. When the tracing point has moved from T to T' , the point O has moved through the arc OO' and the tracing point subtends in its

¹ In the mathematical discussion following, the graduated wheel will be considered as if it were a part of the arm TO , with its plane exactly at right angles to the axis of this arm.

movement an angle WPW' , which is equal to the angle TPT' , marked e , which was passed over by T . Then the following relation is easily obtained:

$$WW' = PW' \times c.$$

In this equation the symbol c is a constant, expressing the ratio (for a given angle WPW') of the length of an arc to the corresponding radius for any value of this radius. In other words, in terms of the calculus, this constant would be expressed in **radians**. In general, for every angle there is a constant value which when multiplied by the radius gives the length of the arc for that radius.

The component of WW' corresponding to the **rolling** of the wheel is WX , which is approximately equal to the arc WW' times $\cos W'WX$. That is,

$$WX = PW' \times c \times \cos W'WX. \quad (8)$$

But if PY is drawn perpendicular to $T'W'$ produced

$$PW' \cos W'WX = W'Y, \quad (9)$$

and combining Eqs. (8) and (9),

$$W'Y = \frac{WX}{c}. \quad (10)$$

Since the angle WPW' is very small, WW' may be taken as being perpendicular to $W'P$. Now WX is perpendicular to $T'Y$ and the angle $W'WX$ is equal to the angle $PW'Y$. The trigonometric relations reducing the foregoing to terms of the length of one of the arms of the planimeter and the constant angle a are as follows:

$$\begin{aligned} W'Y = \frac{WX}{c} &= PW' \cos PW'Y = PO' \cos PO'Y - W'O' \\ &= PO' \cos a - W'O', \end{aligned}$$

then

$$WX = c(PO' \cos a - W'O'). \quad (11)$$

This is an expression for the amount of **rolling** of the wheel when the tracing point moves from T to T' .

To express the relations required, the area will now be expressed trigonometrically in similar units. From geometry, the area of sector $TPT' = \frac{1}{2}$ arc $TT' \times PT$, but arc $TT' = PT \times c$, or area $TPT' = \frac{1}{2}c \times \overline{PT}^2$.

We can write, also,

$$\begin{aligned} PT &= \sqrt{\overline{PO}^2 + \overline{OT}^2 + 2PO \times OT \cos a}, \\ \text{area } TPT' &= \frac{1}{2}c(\overline{PO}^2 + \overline{OT}^2 + 2PO \times OT \cos a). \end{aligned} \quad (12)$$

But the area represented by the amount of rolling of the graduated wheel is that part of the sector **outside** the zero circle (see page 102), and this is

the area $TT'Q'Q$. Now the radius r of the zero circle, referring again to Fig. 82,¹ is easily obtained from equations expressing the relations of the sides of the right triangles in that figure for the particular case when there can be no rolling movement. Thus,

$$\overline{PO^2} = \overline{WO^2} + \overline{PW^2}, \quad (13)$$

$$\overline{PW^2} = \overline{PT^2} - \overline{WT^2} = \overline{PT^2} - \overline{WO^2} - 2WO \times OT - \overline{OT^2}. \quad (14)$$

Combining Eqs. (13) and (14),

$$\overline{PO^2} = \overline{WO^2} + \overline{PT^2} - \overline{WO^2} - 2WO \times OT - \overline{OT^2}.$$

But $PT = r$, the radius of the zero circle, therefore,

$$r = \sqrt{\overline{PO^2} + 2WO \times OT + \overline{OT^2}}. \quad (15)$$

Also from geometry, as explained on the preceding page,

$$\begin{aligned} \text{Area } QPQ' &= \frac{1}{2}r \times c \times r = \frac{1}{2}c \times r^2 \\ &= \frac{1}{2}c(\overline{PO^2} + 2WO \times OT + \overline{OT^2}). \end{aligned} \quad (16)$$

Subtracting Eq. (16) from Eq. (12),

$$\text{Area } QTT'Q' = c \times OT(PO \cos a - WO). \quad (17)$$

Equation (17), which is the expression for the area outside the zero circle, will be observed to be **equivalent** to the roll of the graduated wheel, as given in Eq. (11), times the length of the arm OT from the pivot to the tracing point. If, therefore, for a given area A , we call the reading of the wheel R and the length of the arm from pivot to tracing point L , then

$$A = LR. \quad (18)$$

It should be noted further that this equation is independent of any other dimensions of the instrument.

That this demonstration applies to areas not adjacent to the zero circle or partly inside and out can be readily shown by subtracting in a given case the area between the zero circle and the required area.

Area of Zero Circle by Experiment.—The area of the zero circle of a planimeter may be found readily by passing the tracing point around the circumference of two circles each larger than the zero circle. Preferably for this operation the fixed point of the instrument is placed at the center of the circles. If the calculated areas of these circles are respectively A_1 and A_2 , and r is the radius of the zero circle, then, since readings of the graduated wheel show only the areas outside the zero circle represented by R_1 and R_2 , we obtain

¹ It will be remembered that with the arms of the planimeter in the position shown in Fig. 82 the tracing point T describes the circumference of the zero circle.

$$\begin{aligned}
 A_1 &= \pi r^2 + R_1, \\
 A_2 &= \pi r^2 + R_2, \\
 2\pi r^2 &= A_1 + A_2 - (R_1 + R_2).
 \end{aligned}
 \tag{19}$$

After r has been found,¹ it is not difficult to calculate the proper length of the arm OT for any linear units (compare Eq. (15)). In fact, very many polar planimeters are constructed with the arm OT adjustable, so that the instrument can be used for any scale or for various units. The exact lengths required for both the English and metric units (inches and centimeters) are usually stamped on the adjustable arm.

Mean Ordinate of an Area.²—If we call m the mean ordinate and l the length of a given area A , then

$$A = ml.$$

From Eq. (18) we have

$$A = LR,$$

whence

$$\begin{aligned}
 ml &= LR, \\
 m &= \frac{L}{l}R.
 \end{aligned}
 \tag{20}$$

When, therefore, the tracing-point arm is adjustable, it may be set as shown in Fig. 84* to make it equal to the length of the area measured. Then, obviously, the height of the mean ordinate will be equal to the reading of the graduated wheel expressed in the same units. For example, if the subdivisions of the wheel are fortieths of an inch, the result will be the mean ordinate, also in fortieths. This scale of the wheel is

¹ If, instead of measuring and calculating the circles both larger than the zero circle, one of the two is made smaller than the zero circle, then the reading of the instrument is again the difference between the area of the circle and that of the zero circle, but the value of this difference is now negative, so that, if A_1 is the area of the circle larger than the zero circle and A_2 is the area of the one smaller, then, with the other symbols as before,

$$\begin{aligned}
 A_1 &= \pi r^2 + R_1, \\
 A_2 &= \pi r^2 - R_2, \\
 2\pi r^2 &= A_1 + A_2 - (R_1 - R_2).
 \end{aligned}$$

Although this latter method does not fall in with the general demonstration so well, it is, however, usually preferred, as it will give greater accuracy than can be obtained with two circles both larger than the zero circle, unless one of these is made unusually large.

² Engineers must calculate **mean ordinates** most often when determining the **mean effective pressure** (m.e.p.) of engine indicator diagrams.

* To facilitate the adjustment of the arm to the length of the diagram or area measured, sharp points M and N are attached to the back of some planimeters. The point M is often conveniently placed a short distance away from the tracing point T , and the point N must then be the same distance and in the same direction away from the pivot O . Then, obviously, the distance between M and N will be in all cases equal to the length of the adjustable arm.

determined not by the diameter of the portion of the wheel which is graduated, but by the diameter of the **edge** which comes into contact with the surface over which the wheel rolls. If, then, d is the so-called diameter of "rolling" of the wheel, its circumference is πd . Now by

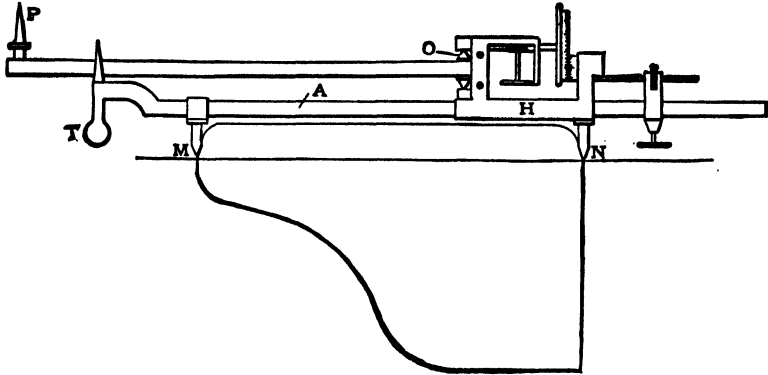


FIG. 84.—Polar planimeter with adjustable arm for the rapid determination of mean ordinates.

dividing the number of divisions on the circumference (usually 100) by πd , the "scale" of the wheel is obtained. It may also be found by measuring a rectangular area of the same length as that of the **tracer** arm and 1 inch wide, when the reading from the wheel will give the

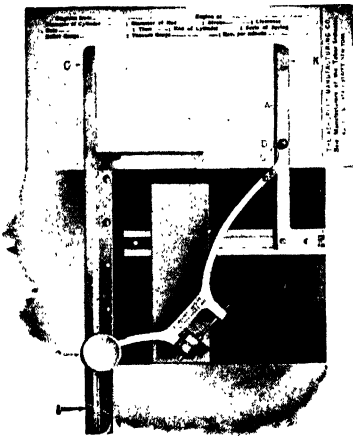


FIG. 85.—Coffin planimeter.

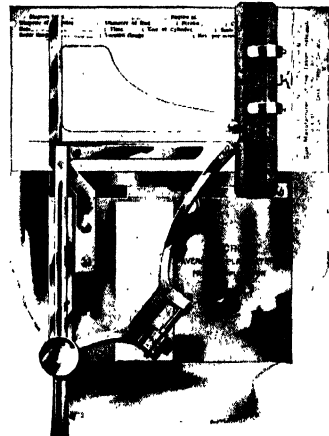


FIG. 86.—Coffin-Ashcroft averaging planimeter.

number of divisions per inch. For those instruments of which the radius of the wheel is 1 centimeter (0.795-inch diameter) and has 100 divisions, the scale is almost exactly 40 divisions to the inch.

Coffin Planimeter and Averaging Instrument.—This planimeter is made commonly in two forms, illustrated by Figs. 85 and 86. The

former is somewhat the simpler and will be explained first. In principle the two are exactly alike. As will be observed in the figures, this instrument has a single arm, to which a suitably graduated wheel is attached on an axis parallel to the line joining the ends of the arm. One of the ends of this arm is for tracing the outline of the area measured while the other slides up and down in a suitable slot. One of the advantages of this instrument over the polar planimeter, although it is not so generally adaptable, is that the wheel is made to move over a specially prepared surface, preventing unnecessary slipping. On materials having a rough, fibrous, or, worst of all, uneven surface, the movement of the wheel of any planimeter will not be the same as when rolling over a smooth flat surface.

The Coffin planimeter may be discussed as a special form of the general polar type in which the pivoting point O , instead of swinging about the

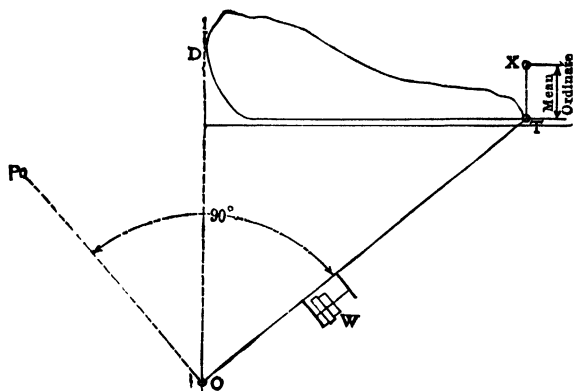


FIG. 87.—Theoretical diagram for a Coffin planimeter.

fixed point P (Fig. 80), moves back and forth in a straight line. The angle between the arms PO and OT , as indicated by the dotted lines in Fig. 87, is really invariable at 90 degrees. Obviously, then, Eq. (17) expressing the area traced by a polar planimeter outside the zero circle becomes, referring to Fig. 83,

$$\text{Area} = c \times OT(-WO);$$

likewise Eq. (11), expressing the roll of the wheel for the Coffin planimeter, becomes equivalent to

$$\text{Roll or record of the wheel} = c(-W'O') = c(-WO).$$

Using, as before, in Eq. (18), the symbols L and R for, respectively, the length of the arm OT and the reading of the wheel, we have, just as for the polar planimeter,

$$A = LR. \quad (21)$$

As an **averaging instrument** the Coffin planimeter is very much more convenient than the typical forms of polar planimeters. For finding the **mean ordinate** of an area, the use of the polar type of these instruments was explained on page 105. The sliding vertical straightedge shown at the right in Figs. 85 and 86 is for the purpose of making the operation of finding the mean ordinate of an area (or the "mean effective pressure" of an engine indicator diagram) as simple as possible. For this operation the straightedges *C* and *K* should be adjusted so that when the tracing pin passes over the extreme end of the area to be measured it will just touch both of them. Now if the tracer is started at either **end** of the area and moved around to the starting point and then moved upward along the vertical straightedge until the reading of the wheel is the same as when starting to trace the area, this last distance traced from the starting point along the vertical straightedge is the **mean ordinate**. To demonstrate this statement, the symbols used on page 105 will be continued. Representing the mean ordinate by *m*, the length of the area *A* by *l*, the reading or rolling of the graduated wheel in **going around the area** by *R*, and the length of the arm carrying the tracer by *L*, then as before

$$A = ml.$$

Now, when the tracing point *T* moves over a vertical line, the angle *DOT*, represented by *Z* in Fig. 88, remains constant. If we call the vertical distance moved *V* and remember that only the movement of the wheel at right angles to its axis produces rolling, then the reading *R* corresponding to the rolling is

$$R = V \sin Z. \quad (22)$$

But for the position shown in Fig 87. when the tracer *T* is at the right-hand end of the outline of the area, we have

$$\sin Z = \frac{l}{L},$$

whence

$$R = \frac{Vl}{L}.$$

Substituting this value of *R* in the general Eq. (20) for the mean ordinate *m* of a polar planimeter, then,

$$m = \frac{L}{l} \frac{Vl}{L} = V. \quad (23)$$

This relation can be illustrated more simply, however, by referring to Fig. 89, which is a typical indicator diagram from a steam engine. In this figure the tracing point of the Coffin instrument is shown at *O*, with

the tracing arm represented by VO . A rectangle $OXYZ$, indicated by dotted lines, is shown, of which the area is equal to that of the indicator diagram. Starting at O and moving the tracing point around the **indicator diagram** once, the difference in readings is the area. Now if the tracing point is moved in the opposite direction around the **rectangle** and again back to the starting point at O , it will measure a negative area equal to the first area and the reading of the graduated wheel will be the same as when first started around the indicator diagram. The

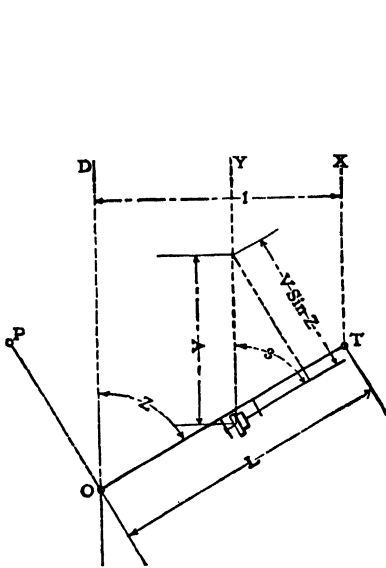


FIG. 88.—Theoretical diagram for a Coffin planimeter.

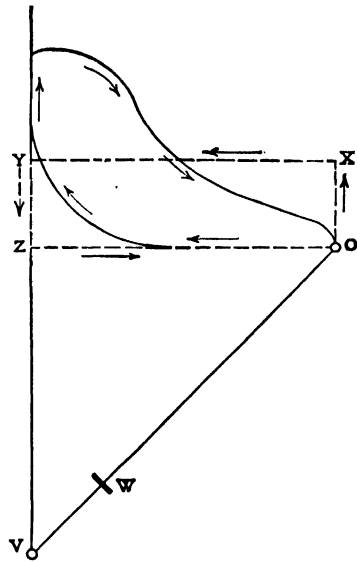


FIG. 89.—Diagram explaining the method of mean ordinates with a Coffin planimeter.

movement of the graduated wheel as the tracing point moves from X to Y is equal and opposite to that in going from Z to O , so that these two cancel each other. The motion of the tracing point from Y to Z requires the axis of the graduated wheel to be parallel to YV and consequently during this movement the wheel will not be moved. The only movement that is therefore producing a net change in reading of the graduated wheel during the reverse tracing of the rectangle is in going from O to X . Consequently, after going around any irregular area like an indicator diagram in a clockwise direction from the starting point at O at the right-hand end of diagram, if the tracing point is moved in a vertical direction from the starting point at O until the reading of the graduated wheel is the same as when first started, this vertical distance moved, measured from O , will be equal to the mean height of the indicator diagram.

Although measurements of areas may be made with the Coffin planimeter as with the regular polar types with the area in any position as

regards its length and breadth, yet when the mean ordinate is to be obtained, its value in a definite position is required and the area must be placed so that its length with respect to which the mean ordinate is to be obtained will lie along the horizontal straightedge shown in the figures. Then the mean ordinate measured along a vertical straightedge will give the result required.

Roller Planimeters.—For the measuring of very large areas, a planimeter differing slightly in theory from the polar type has been designed by G. Coradi, of Zurich, Switzerland. It has the advantage of being

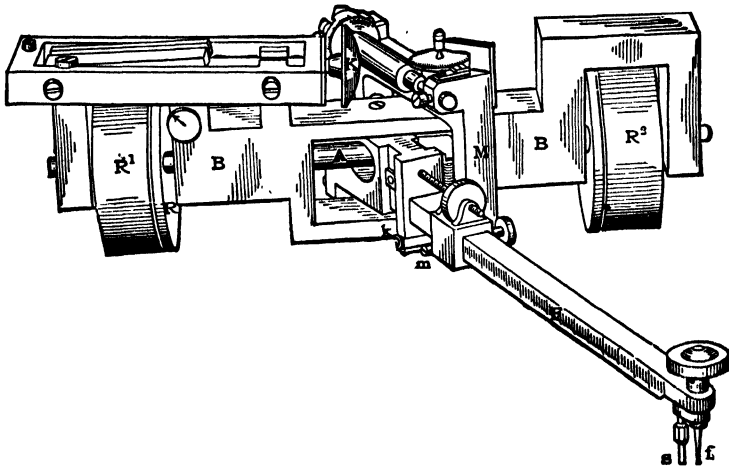


FIG. 90.—A typical roller planimeter.

adaptable for measuring surfaces of indefinite length and as wide as the length of the tracer arm. This instrument is illustrated in Fig. 90. It is supported at three points—the two rollers R^1 and R^2 and the tracing pin f , or its support s . These two rollers are attached to the shaft A . On the **face** of one of these rollers is a minutely divided miter wheel engaging with a small pinion revolving the horizontal shaft carrying the spherical segment K . At the center of the frame B , and in the same vertical plane with the two shafts already mentioned, a vertical shaft carrying the tracer arm is supported. The spherical segment K causes merely by friction contact the movement of the cylindrical “measuring” roller shown at its right. This roller is supported on the auxiliary frame M , of which the tracer arm is a part. The “measuring” roller moves back and forth with respect to the spherical segment to correspond with the movement of the tracing point; but at the same time the rotation of the segment itself imparts rolling motion of the entire instrument.¹

¹ Since this instrument is not often used by engineers, those interested in its theory are referred to Coradi's book of directions (in English) accompanying each instrument, or to “Handbuch der Vermessungskunde,” by W. Caville.

Calibration of Planimeters.—Tests are made by comparing the readings of the instrument with that calculated for a given area. For such calibrations it is necessary to use an area which can be gone over accurately with the tracing point preferably held mechanically. This is done usually by using a metallic testing rule, shown in Fig. 91. It is usually

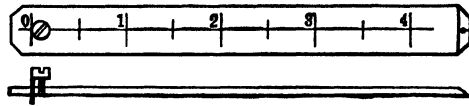


FIG. 91.—Planimeter testing rule.

made in the shape of a narrow strip from 3 to 5 inches long. At the end marked zero on the graduations a needle point is set which is kept in place by an overlapping screw. At each line of the graduations there is a very small conical hole into which the tracing point of the planimeter can be placed. The beveled end of the testing rule has the index line set accurately at the starting mark, so that this point can be very carefully located. With the tracing point *T* in the testing rule and the fixed point *P* of the planimeter in approximately the position shown in Fig. 92, observe the reading of the instrument corresponding to the area of the circle described by the tracer moving clockwise, in the positions shown,

1. When the fixed point *P* is on the left-hand side of the tracing point.
2. When *P* is on the right-hand side.

If the reading obtained is greater in the first position than in the second, the end of the shaft carrying the graduated wheel nearest the tracing point must be shifted toward the right to make the instrument accurate, and *vice versa*. Otherwise the error, if there is one, can be eliminated by taking the mean of the results obtained for the two positions.¹

Another test to be made, if there is doubt about the accuracy of a planimeter after the axis of the wheel has been adjusted, is to determine whether the settings of the adjustable arm marked on the instrument are correct. For this determination, circles with several different diameters can be measured with the testing rule; and if there is a nearly constant

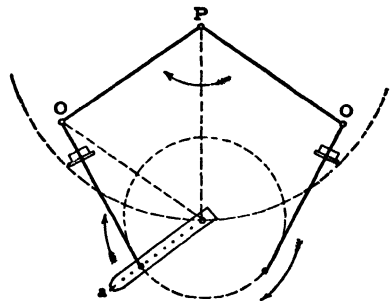


FIG. 92.—Methods of testing planimeters.

¹ For ordinary requirements a testing disk can be used in place of the rule, although it is not usually so accurate. On this disk circles of 1, 2, and $2\frac{1}{2}$ inches diameter are usually engraved; and if neither a testing plate nor a disk is available, tests can be made by using circles drawn with a pencil compass on a flat sheet of well-calendered paper.

percentage error, say x per cent too large, then the adjustable arm must be lengthened x per cent to make the planimeter readings correct, and *vice versa*.

For accurate results the fixed point P should be placed as indicated by the dotted lines in Fig. 92, so that, when the tracing point is near the center of the area to be measured, the two arms will be approximately at right angles.

Bristol-Durand Integrating Instrument.—This unusual instrument, illustrated in Fig. 93, has been developed by the Bristol Company for

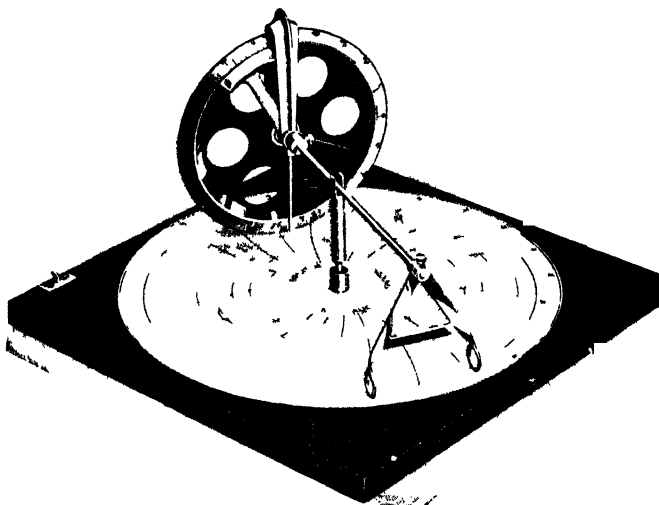


FIG. 93.—Bristol-Durand integrating instrument for circular charts.

obtaining the average radius of records traced on circular charts of *uniform graduations* like those used in recording gages, thermometers, etc. It is a simple device for obtaining quickly the average value of pressure, temperature, draft, watts, volts, amperes, and other records generally taken on circular charts.

This instrument consists of a wooden base in which there is a metal socket for supporting a rotatable pin slotted for receiving a horizontal shaft to which the integrating wheel is rigidly attached. On this shaft between the integrating wheel and the pin there is an adjustable tracing point and at the opposite end of the shaft there is a triangular support for the shaft, also adjustable.

The general principle of this instrument was evolved by Prof. W. F. Durand¹ of Leland Stanford University. Its application hinges on the condition that the chart to be measured has a uniform radial scale, just as there must be a uniform vertical scale for indicator and other similar diagrams in order that they may be averaged with ordinary planimeters.

¹ *Trans. A.S.M.E.*, Vol. 29, 1908.

Obviously, the mean value of the radius of a circular diagram cannot be determined with ordinary planimeters, since the area of a diagram in polar coordinates is proportional to the square of the radius and to the angle.¹ In Fig. 94, AB is an irregular curve, considered for this theoretical discussion as traced by a point moving in and out on a **straight radial line**. The center of the chart is at O , and at this point there is a socket, in which a rod $O'P$ slides freely back and forth, permitting a tracing point P to draw a curve AB . A graduated wheel W attached to $O'P$ serves the same general purpose as the integrating wheel in the ordinary planimeter. Obviously, this wheel will be moved only by

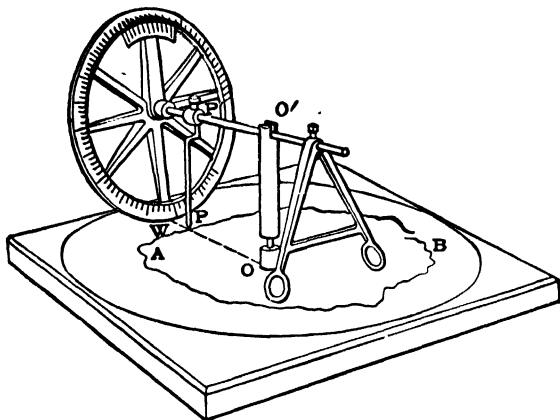


FIG. 94.—Diagrammatic drawing of Bristol-Durand integrating instrument.

circumferential motion, and for any radial movement of the rod in the direction of its length it will remain stationary. The amount of movement will be proportional to the radius WO , which differs from PO by a constant distance WP . The resultant movement of the wheel W is proportional, therefore, to the angle moved by the arm $O'P$ and to the radius OW varying from point to point along the curve. If it is assumed for the present, that, as will be shown later, the reading for any part of the curve, as AB , is proportional to the product of the angle subtended between the points A and B , AOB , and the mean radius for the curve between these points, then if this reading is divided by the subtended angle expressed in circular measure, the quotient will be proportional to the mean radius. Now if to the value of this mean radius the constant distance WP is added, the true value of the radial ordinate OP is obtained. When, as is usually the case in practice, the curve AB represents values of radial ordinates with reference to a base circle of constant radius as the datum or "zero line," then if the radius of this base circle is

¹ With the ordinary planimeter the mean square of the radial ordinates can be determined, and we can, of course, take the square root of these values, but in most cases this is not the same as the mean radius.

subtracted from OP , the remainder will be the true value of the ordinate. By making WP equal to the radius of the base circle, as may readily be done by a suitable adjustment of the instrument, the two corrections will be "balanced" and the mean value of the radial ordinate will be given directly as the quotient of the reading of the wheel and the subtended angle AOB expressed in circular measure. For a chart corresponding to 24 hours for a circumference, the angular measure to be used as the divisor will be 0.2618 per hour.

The quantity to be determined in such diagrams is the time mean of the quantity measured by the radial ordinate. But since angular motion is made proportional to time, we may represent the desired mean by the following integral formula:

$$r = \frac{\int r \delta \theta}{\int \delta \theta} = \frac{\int r \delta \theta}{\theta}. \quad (24)$$

Now, in Fig. 95, let $ABCD$ denote a curve drawn by a tracing point which moves on the arc of a curve shown by OAV instead of on a straight radial line. Then let OV , ON , OM , etc., denote a series of consecutive positions of the curve OAV , at differential angular intervals $\delta \theta$. Then for the actual curved path $ABCD$ substitute the broken-line path made up of a series of arcs each $r \delta \theta$ in length, and the series of differential bits of the curve OAV as shown. Then **at the limit** the record of any integrating or averaging instrument will be the same, whether the tracing point is carried along the curve or along the broken line as shown.

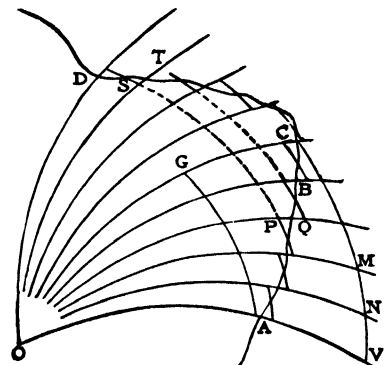


FIG. 95.—Theoretical curves for Bristol-Durand instrument.

Then suppose an integrating instrument, as shown in Figs. 93 and 94, is applied to such a diagram, and let the tracing point P be carried along the zigzag path. The record of the wheel will be made up of two parts:

1. That due to the circular arcs $r \delta \theta$ and representing by summation the value of $\int r \delta \theta$.
2. That due to the differential portions of the arc OAV .

Now it is clear that, if the diagram extends all the way around from A through BCD to A again, the differential elements of the curve OAV may be considered as existing in pairs, and that for every element traversed in the **outward** direction there will be an equal element traversed in the **inward** direction. PQ and ST denote the members of such a pair. The record for such a pair will therefore disappear in the summation—that is, for all the pairs, and also for the diagram as a whole. In such

a case, therefore, part 2 above becomes zero and the record of the wheel for the entire diagram consists simply of $\int r\delta\theta$.

This reasoning is seen to be entirely general and independent of the character of the path OAV and hence must be true whether it is the arc of a circle, a straight line, or any other path.

In case the curve occupies only part of the revolution, as ABC , then it is clear that in going from A to C the record will involve the two parts 1 and 2 above, and that the latter will remain included in the final result and will represent the summation of the record due to the elements of OAV between A and C . This, obviously, will be the value of $\int r\delta\theta$ for the arc GC and it will be canceled by carrying the tracing point of the instrument back from C to G . This method of reasoning is **independent of the extent of the arc and is therefore equally true for an entire revolution, even when the diagram does not end at the same radial distance, as at the beginning.** In such cases it is necessary only to trace along the arc OAV so as to "close" the curve, thus canceling part 2 above and finding directly the value of $\int r\delta\theta$ for a whole revolution.

In all cases the correction for part 2 of the record is made by tracing from the terminal point of the curve along the path, representing no change of time, to a point lying in a circumference passing through the initial point. This may be stated in other words by saying that to eliminate part 2 of the record the tracing point must start and finish at the same distance from the center, and, if the diagram is not of the kind to satisfy this condition, then the necessary portion of a path of zero change of time must be used to supplement the diagram. This discussion is independent also of the nature of the curve OAV . It may be stated, however, that when OAV becomes a straight line the value of the correction becomes zero.

Planimeters for the same purpose are made by Builders Iron Foundry, Foxboro Company, and Keuffel & Esser Company.

CHAPTER V

ENGINE INDICATORS AND REDUCING MOTIONS

The engine indicator is simply an instrument showing by graphic diagrams the variations of the pressure in the engine cylinder of steam, gas, air, or whatever the working substance may be. Before James Watt invented the engine **indicator** (about 1814) he had already used a steam

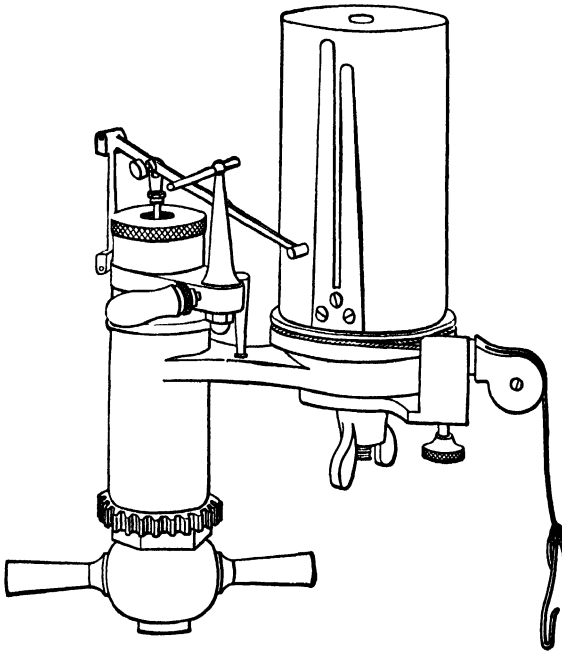


FIG. 96.—Typical Thompson indicator.

pressure gage on the cylinder of his engine, and, since the movement of the piston in the early steam engines was very slow, he was able to observe with his eyes how the pressure varied during a stroke of the piston. In modern engines the movement of the piston is so rapid, however, that a recording instrument is absolutely necessary.

Thompson Indicator.—Of the engine indicators now in general use the Thompson is the oldest and best known. Figure 96 shows one view of this instrument and Fig. 97 shows the corresponding sectional drawing.

It consists in essential parts of a piston 8 (Fig. 97) moving in a cylinder 4. This piston is rigidly connected to the rod 12, which passes up

through the cap 2. The motion of the piston rod 12 is transferred to the pencil 23 by means of suitable links designed to make the pencil move parallel to, but usually four times as far as, the piston 8. The maximum pressure of the pencil on the paper used for the diagram is adjusted by the thread and set screw on the handle attached to the bracket X.

The method of changing the springs in the various common forms of engine indicators should be well understood by everyone likely to be called on to "indicate" engines.

When the work of changing springs is done clumsily or carelessly, a great deal of time is often wasted by the whole party engaged in the test. The method to be followed in changing springs of a Thompson indicator may be stated briefly as follows: The milled-edge cap 2 should first be unscrewed from the top of the cylinder containing the spring and piston. This cap, together with the sleeve and bracket X carrying the pencil lever and linkages, the piston rod, and the piston, can then be lifted from the main body of the indicator. By unscrewing the small milled-headed screw 19 connecting the piston rod with

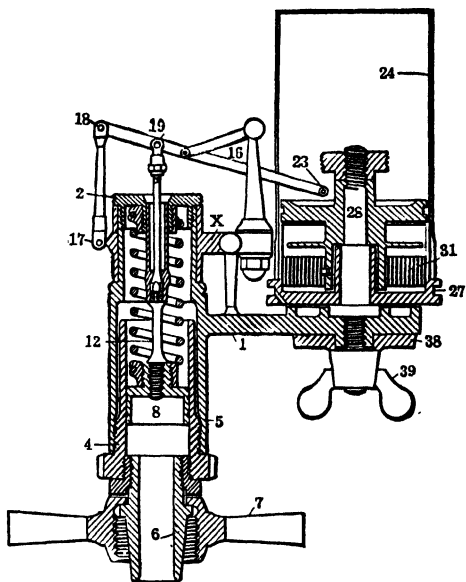


FIG. 97.—Section of Thompson indicator.

the pencil arm, the spring can then be unscrewed, first from the cap 2 and finally from the piston 8. By exactly reversing the operation, another spring can be put in the place of the one removed. Changing springs in this instrument is a simple operation. No wrenches or other tools are required. Care should be taken, of course, to screw up the spring firmly against both the cap and the piston. Probably one-half the troubles with indicators in operation arise from loose springs, although not so often, probably, with Thompson indicators as with some other types. The height of the pencil can be adjusted by turning the screw head 19 up or down on the piston rod.

As a general rule, the spring selected for an indicator should be of such a scale that the largest diagram to be taken will not be more than $1\frac{3}{4}$ inches high; that is, if the maximum pressure will be about 140 pounds per square inch, a spring with a scale of 80 pounds should be selected. Instruction books going with indicators have usually tables showing the spring recommended for a given maximum pressure. Generally a higher

card is permissible for light springs and slow engine speeds than for stiff springs and high speeds. The tension of the spring 31 inside the drum carrying the paper for the diagram is varied by loosening the thumb nut and turning the large milled cap until the proper adjustment is secured.¹

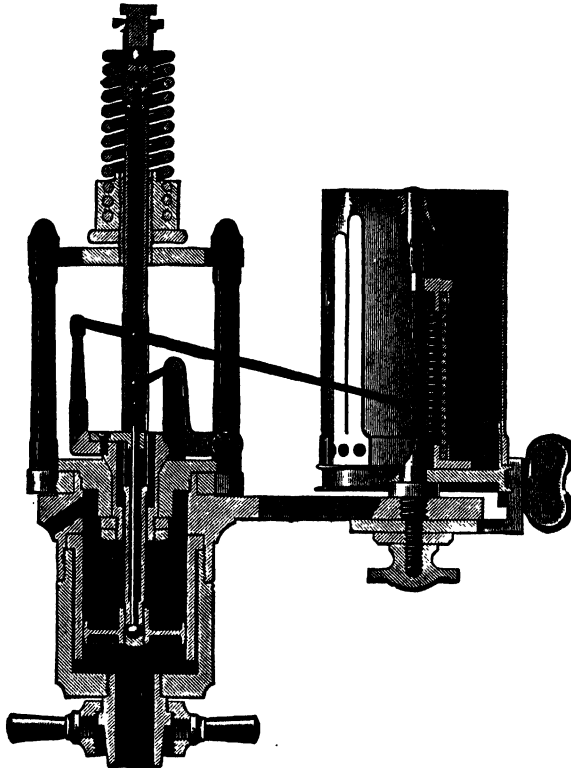


FIG. 98.—“Tension” type of Crosby outside-spring indicator.

Crosby Indicators.—For high-speed engines and for accurate results the Crosby indicator has long been a favorite with engineers. The most satisfactory indicator of this make is shown in Fig. 98.

Indicator springs arranged outside the cylinder as shown here are not subjected to high temperatures and are particularly desirable for use with engines operating with superheated steam. There are two principal advantages: (1) The spring can be changed without removing the piston, which avoids an operation often causing confusion and loss of

¹ Unless there is a very good reason for a change in the tension of the spring in the drum, it should not be altered. Too much tension in the drum spring will excessively stretch the cord and produce inaccurate diagrams (p. 150). Particularly in indicators which have been used a long time the pin holding the spring in place is likely to be much worn, so that if adjusted often the spring may get loose, and then there is usually considerable difficulty in getting it again into its proper position.

time; (2) the tension of the spring cannot be affected by exposure to very high temperatures. The spring can be changed when the thumbscrew at the top of the central spindle has been unscrewed.

Crosby indicators are also made with the spring inside the cylinder of the indicator. The adjustment of this type is much more difficult than the outside-spring type and will therefore be explained in detail.

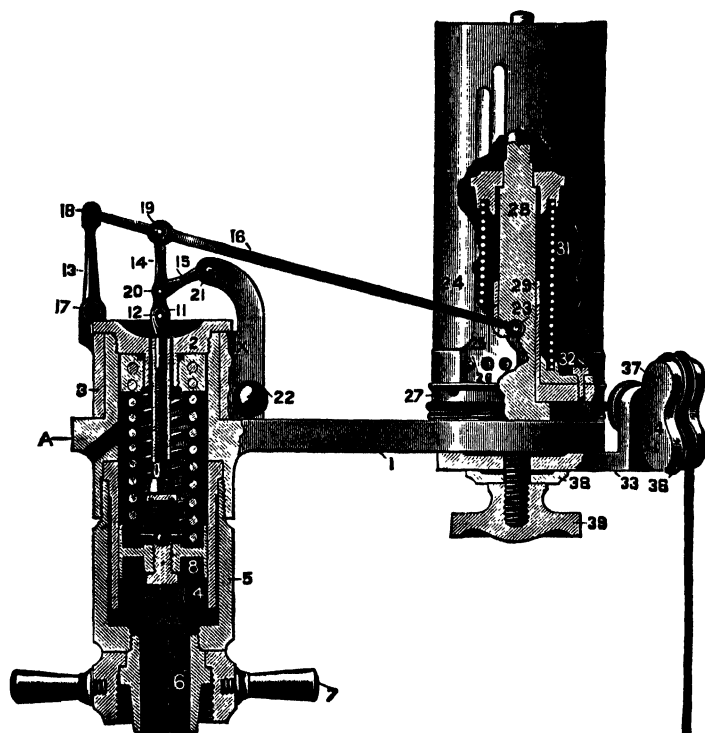


FIG. 99.—Section of Crosby indicator (inside-spring type).

In essential parts the Crosby *inside-spring* indicator (shown in Fig. 99) consists of a piston 8 moving in the cylinder 4 and is connected by means of the piston rod 10 and the link 14 to the pencil lever 16. All of the pencil mechanism (arranged to move the pencil point 23 in a straight line parallel to the motion of the piston 8) is supported by the links 13 and 15 on the sleeve 3. The indicator spring is fastened at its lower end to the piston by a ball joint and at its upper end it is screwed into the cap 2. The method of attachment of the spring to the piston by means of the ball joint is shown more in detail in Fig. 100.

In this indicator the spring is changed by first unscrewing the milled cap 2, then this cap, the sleeve 3, the piston rod 10, and the connected parts can be removed from the cylinder 4. By unscrewing the spring, by hand from the cap, which, of course, must be prevented from turn-

ing, and also from the screw on the swivel head 12, the piston, the spring, and the hollow piston rod 10 are detached from the other parts. A socket wrench of the special form provided in every indicator box of this make is to be slipped over the piston rod to engage with the small nut shown (at 10) in Fig. 99 at the lower end of the piston rod. Then the piston rod is readily unscrewed from the piston and at the same time the spring is released from its attachment to the piston. Now with the piston rod still in the socket of the wrench, slip the spring to be used over the piston rod until the head of the spring rests in the concave end of the rod. To do this, the

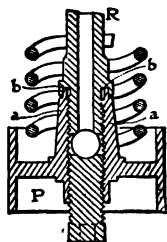


FIG. 100.—Section of Crosby indicator spring and piston.

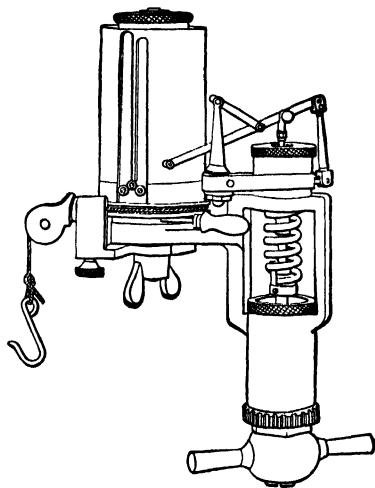


FIG. 101.—“Compression” type of outside-spring indicator.

wrench must be held upright, and then, if the piston is inverted or, in other words, if it is held so that the end screwing into the piston rod points downward, the piston rod is ready to be screwed into the piston, so that the transverse wire of the spring passing through the bead will be held firmly in the slotted portion of the **socket** in the piston. Finally, screw the piston rod firmly¹ into place. Before the last operation, the lower piston screw (Fig. 100) should be loosened slightly, and afterward it should be screwed up against the bead just enough to prevent lost motion. It should not be screwed so tightly, however, as to prevent the bead from turning, otherwise the desirable qualities of the ball joint for securing perfect alignment are lost. Now when the piston rod, spring, and piston are again assembled, if the sleeve 3 and the pencil motion attached to it are held in an upright position, the hollow piston rod can be slipped over the threaded portion of the swivel head 12 until the threads on the upper end of the spring engage with those on the cap 2. The spring can then be screwed securely into the cap 2. Then permit the cap to turn in the sleeve 3 and, by still turn-

¹ Special care should be taken when putting a spring into a Crosby indicator that the piston rod is screwed into its socket in the piston *P* (Fig. 100) as far as it will go, that is, until the extreme upper end of the socket *aa* is brought firmly against the bottom of the corresponding annular channel *bb* in the piston rod *R*.

ing the spring, screw the piston rod on the swivel head 12 until the top of the rod is nearly flush with the shoulder on the swivel head. The piston and attached spring are now ready to be put into the cylinder by slipping the sleeve 3 into position and screwing down firmly the cap 2.¹

The height of the pencil cannot be adjusted to change the position of the atmospheric line without removing the piston from the cylinder of the indicator. It must be done, however, by unscrewing the cap 2 from the cylinder and removing it together with the sleeve 3 and the pencil mechanism. By turning the cap clockwise the swivel head 12 and consequently also the atmospheric line **are lowered**. By turning in the opposite direction both are raised. Never try to make adjustments by removing or loosening the pins or screws at the joints 17, 18, 19, 20, and 21. These joints should always be kept tight enough to prevent any lost motion, and occasionally they should be lubricated with refined porpoise oil of the kind usually supplied with indicators.²

The tension of the spring **in the drum** is changed very much more conveniently than in most other indicators. For high-speed engines the tension must be considerably greater than that required for those running at low speeds. The tension is adjusted by removing the drum 24 by a straight pull and turning the knurled nut at the top of the spring 31 after lifting it from its square seat.

In Fig. 101 a slightly different outside-spring arrangement is shown. It is distinguished particularly from Fig. 98 in having the indicator spring in compression instead of being in tension as in most other outside-spring types. The obvious advantages of the designs having the springs in tension are that springs can be changed much more readily than in other types, and that it is practically impossible if the springs and piston are well made for the spring to buckle over and bind the piston, as happens frequently in all types having the spring in compression.

The weakness of the designs having the spring in tension is in requiring a very long and slender piston rod which, being in compression, may

¹ Persons in charge of tests should always inspect indicators before the steam pressure is put on the springs to observe whether the cap has been screwed down firmly, and whether the pencil mechanism has been adjusted so as to give with a suitable spring a diagram of the proper height.

² Inexperienced testers often put the spring and piston into place by merely slipping on the sleeve 3 and without screwing down the cap 2. Then, as a result, when the steam pressure is put on the indicator the piston, spring, and pencil mechanism are thrown off with a great deal of force, and some of these parts are sometimes completely demolished.

When an indicator having the spring inside the cylinder 4 is used—and this is true particularly in the Crosby indicator—all adjustments should be made before the steam is turned on the indicator, because the piston, spring, and cap soon become very hot, and unless the parts are cooled, preferably by dipping into cold water, they are difficult to handle.

have a tendency to buckle over and produce variable errors. As regards temperature effects, one arrangement is about as good as the other.

Tabor Indicator.¹—In the form in which it is now manufactured, the Tabor indicator differs from indicators like the Crosby particularly in the means employed for producing a straight-line parallel motion for the pencil. In this device a roller is attached to the pencil lever and is arranged to move in curved slots on the inside of the rectangular box-shaped part shown in the figure.

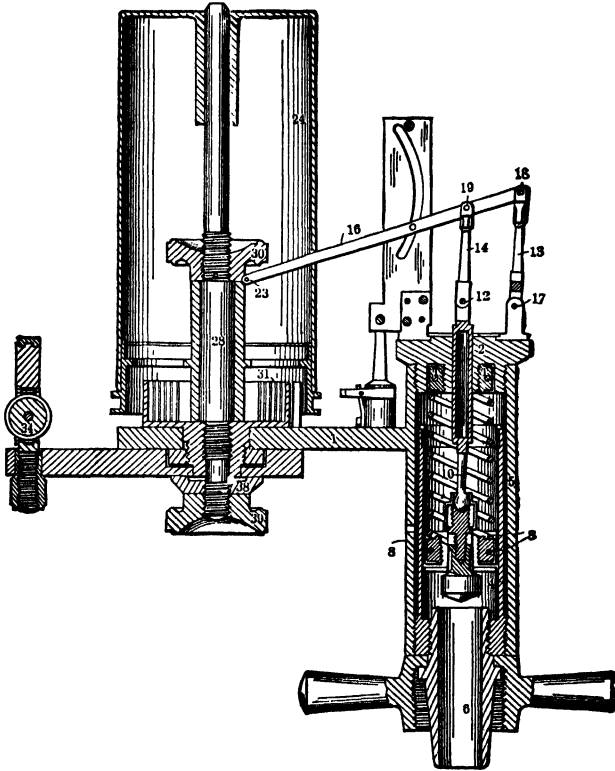


FIG. 102.—Section of a Tabor indicator.

As regards the point of flexibility in the mechanism, this is not between the spring and the piston but, more like the Thompson, is in the ball and socket joint between the piston and the piston rod. Details of this construction are shown in Fig. 102.

The principal precaution to be observed in the use of this indicator is to be certain at all times that the roller on the pencil lever moves freely in the curved slots. Outside-spring types of this indicator are also made.

To Change the Spring.—The cylinder cap must be first unscrewed, and then this cap, together with the piston, spring, and connected parts,

¹ Ashcroft Manufacturing Co., Liberty St., New York.

can be lifted from the cylinder of the indicator. By removing the small screw under the piston the latter can be unscrewed from the lower end of the spring. The other end of the spring can then be unscrewed from the cylinder cap. Another spring is put into the indicator by slipping it over the piston rod with the end stamped *T* uppermost, screwing this end into the cylinder cap, and screwing the piston to the lower end. The pencil mechanism must be moved downward until the piston rod enters the piston and the square shoulder enters the corresponding square socket in the piston. In this last operation care must be taken that the rod is firmly and accurately in the hole, and then the screw at the bottom of the piston should be firmly applied.

Precautions for Care of an Indicator.—Unless an engine indicator is well taken care of, very soon it will be in a condition in which no reliance can be placed on results obtained with it. That the necessary precautions should be taken is all the more important, because it is one of the most expensive as well as most delicate instruments used by an engineer in his ordinary practice. The following precautions are particularly important:

1. Before an indicator is used, all the working parts, especially the piston, should be carefully cleaned. Then after a spring suitable for the pressure has been attached in its proper position and a little cylinder oil has been smeared in a thin coat on the working surface of the piston, the parts should be replaced. Moving parts of the pencil mechanism should be oiled occasionally with watchmaker's or porpoise oil. It is a very good practice, especially when comparatively new indicators are being used on long tests, to take out the piston of the indicator frequently and smear it with cylinder oil. For lubricating this piston it is a little better to use a comparatively thin cylinder oil of high flash test (like gas-engine oil) than one that is very viscous.

2. Adjust the screw on the handle provided for moving the pencil so that when the pencil is sharp the application of the usual pressure on the handle will give a very fine line.

3. Adjust the length of the indicator cord so that the drum will be neither too loose nor too tight or, in other words, so that the drum will not strike either of the stops when the engine is operating. On a small engine this is most easily tested by observing the diagram when the engine is on each of the dead centers. If the diagram is either too long or too short, the drum will not be moved the required distance, and the indicator diagram will be correspondingly too short and therefore inaccurate. The cord used should be selected with care. It must be of such a quality as not to be stretched appreciably by the forces to which it is subjected. For accurate work and on long-stroke engines, fine annealed steel or phosphor-bronze wire, or indicator cord with a wire core,¹

¹ Indicator cord with a wire core which is guaranteed not to stretch in ordinary use can be obtained.

should be used. The length of the cord should be adjusted very carefully and fastened securely so that it will not slip or stretch so as to bring the drum up against one of the stops, making the diagrams too short. This effect can usually be detected by the clicking sound of the drum striking the stop, if there is not too much noise in the room. The experienced engineer, however, will by force of habit invariably put his finger now and then during a test on the top of the drum or on the side of the bracket supporting it to determine whether its operation is satisfactory. Another way to determine a faulty adjustment on the usual crank-shaft type of engine¹ is to measure the lengths of the indicator diagrams. If the cord stretches, the diagrams will be variable in length.

4. The atmospheric line should always be taken preferably after the diagram has been made. It is drawn, of course, when the indicator cock is closed. By this order of procedure in tests, the diagram can be more easily taken exactly "on the signal." The length of the diagram must always be measured on the atmospheric line or on a line parallel to it. The indicator cock should be kept closed and the cord to the reducing motion should be unhooked except when a diagram is to be taken. When the cord is unhooked, the drum should not be permitted to snap back against the stop. By observing these precautions the useful life of an indicator can be much prolonged.

5. **Immediately after a diagram has been taken, it should be removed from the drum and examined.** If there are unusual irregularities in the lines, unaccountable differences in the areas or in the lengths of different cards, the facts should be noted and the best efforts should be made to remedy the faults. Irregularities are usually due to stretching of the indicator cord, grit on the piston, lost motion in the working parts (usually inside the indicator cylinder), or excessive friction caused by overheating of the piston, particularly when used on gas engines.

To correct these faults concerning the piston, it must be removed from the cylinder and should then be carefully cleaned and again lubricated with cylinder oil. Before putting the piston and connected parts back into the indicator cylinder, it should be observed whether or not all the parts are connected firmly and without lost motion.

6. After a test, the indicator should be removed immediately from the engine, protecting the hands with waste or thick gloves to prevent burns. All the parts, especially those in the cylinder, should be thoroughly cleaned and then put together again without the spring, which should be put away with the other springs in a box provided for the indicator. An indicator should never be handled by taking hold of the drum, as usually it is fastened to the indicator by only a loose slip-joint, and this comes off easily.

¹ In an engine without a crank shaft, like a direct-acting steam pump, the length of the stroke, and consequently of the indicator diagram, is likely to be variable.

7. Before opening the indicator cock to take the first diagram in a test, examine the indicator carefully to see that piston, spring, and pencil mechanism are attached securely. A good method is to take hold of the end of the pencil lever near the pencil point and try to move it up and down. If there is no lost motion observable and the pencil point seems to be at about the right height for drawing the atmospheric line, it may be assumed that the indicator has been assembled properly. Otherwise it will be observed immediately by this test if the indicator has been put on the engine without inserting a spring, or if the milled nut at the top of the indicator cylinder has not been firmly screwed down. Observe also whether the "union" nut attaching the indicator to the cock is held by at least three or four good threads. Otherwise if this nut slacks back a little the whole indicator may be thrown by the force of the steam pressure against the ceiling of the room. More indicators are worn out and broken by careless assembling than in any other way.

8. One of the best ways to put an indicator card on the drum is to first bend over one of the short edges of the card on a line about $\frac{1}{4}$ inch from the end and place this end with the line of bending snugly against the top of the longer clip on the drum. Then lap the card around the drum and insert the other end of the card into the upper end of the shorter clip.

The card should then be pushed down to the stops in the clips, being careful, however, to keep it tight and straight, so that there will be no wrinkles. Finally, to prevent the card from shifting on the drum, the end of the card under the shorter clip should be bent over carefully and firmly.

Optical Indicators.—The usual types of indicators operating with a piston are not suitable for engines running at much over 400 revolutions per minute. For higher speeds, optical indicators are used. These operate by the deflection of a beam of light from a mirror, the deflection being proportional at any instant to the pressure. When such a device is used on an engine, successive indicator diagrams can be readily observed and compared by marking with a pencil the reflection upon a ground-glass plate; and if a photographic sensitive plate is exposed to the beam of light in the place of the ground glass, a permanent impression can be taken, showing at any instant the operation of the engine. Optical indicators are practically the only kind that can be used successfully for

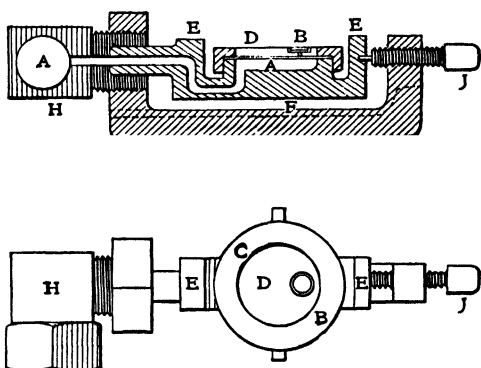


FIG. 103.—Essential parts of a simple optical indicator.

indicating the action of modern high-speed automobile engines. The pressure is communicated to the instrument through the usual type of indicator cock supporting it. A simple reducing motion serves for reducing the length of the stroke of the engine to a suitable size for such a small instrument. A glass mirror moved about an axis by the motion transmitted by the reducing motion from the cross-head and about another axis at right angles to the first, by the pressure in the engine cylinder reflects a beam of light from an electric-light bulb upon a sheet of paper so that the indicator diagram can be traced with a pencil or may be photographed.

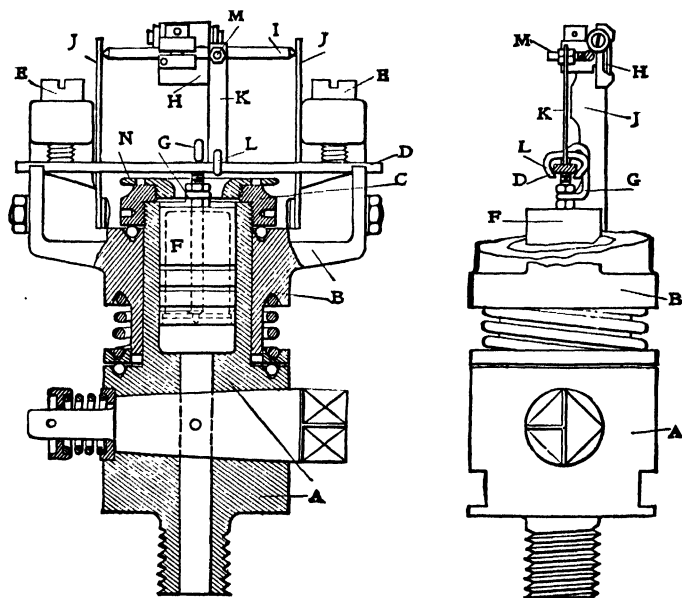


FIG. 104.—Hopkinson optical indicator.

Details of the essential parts of this instrument are shown in Fig. 103. Through the indicator cock the pressure in the engine cylinder is communicated to the cored passages marked *A, A*. This pressure tilts the mirror *B*, attached to the thin steel diaphragm *D*. When, therefore, the mirror is still, a ray of reflected light will be seen as a bright **spot** on the screen; but when moved both by the pressure and by the motion of the cross-head the conventional indicator diagram is traced. It is very interesting to watch the rapid change of shape of such diagrams as load, speed, pressure, cut-off, etc., are changed. With such an instrument these interesting phenomena in engine operation can be illustrated on a ceiling to a large class of students.

The **Hopkinson optical indicator** is shown very clearly in Fig. 104. It is essentially similar to the one in Fig. 103, except that it has a piston *F*

instead of a diaphragm and a direct type of reducing motion is used as shown in Fig. 105.

Newell High-speed Indicator.—A high-speed indicator for automobile engines has been developed by the Bureau of Standards. This device has the advantage over some types that the recording mechanism need not be located close to the engine. This instrument has a thin steel diaphragm (Fig. 106) about an inch in diameter, which is mounted

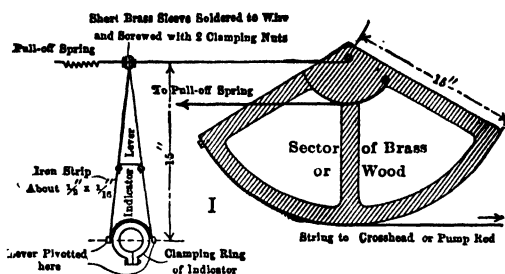


FIG. 105.—Reducing motion for Hopkinson indicator.

between two slightly concave supporting plates which are perforated to permit the pressure to reach the diaphragm. The range of motion of the diaphragm is limited to about $\frac{1}{100}$ inch. The center portion of one of the supporting plates is electrically insulated in such a way that an electric current is made as the diaphragm makes contact with this supporting plate. The body of the instrument is fitted to screw into a spark-plug hole.

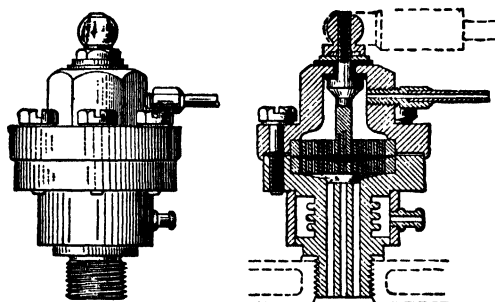


FIG. 106.—Newell high-speed indicator.

A supply of air or gas in a tank under pressures as high as those to be measured is necessary for balancing the engine-cylinder pressures. For the measurement of maximum and minimum pressures occurring in a cycle, a pressure intermediate between the maximum and minimum is applied to the diaphragm from an air or gas storage tank. This pressure is increased until clicking ceases in a telephone receiver in circuit with the electric contact on the diaphragm, for the maximum

pressure, and is decreased for the minimum pressure. To produce a complete diagram of a cycle, a one-point timer is connected in series with the telephone circuit and by setting the timer at a given point the required pressure will be indicated on pressure gages when a clicking sound is heard in the receiver. It is convenient to set the timer at successive positions and adjust the pressures to correspond. An automatic recording mechanism has been devised which makes a photographic record of electric sparks when the required pressures are obtained as shown in Fig. 107.

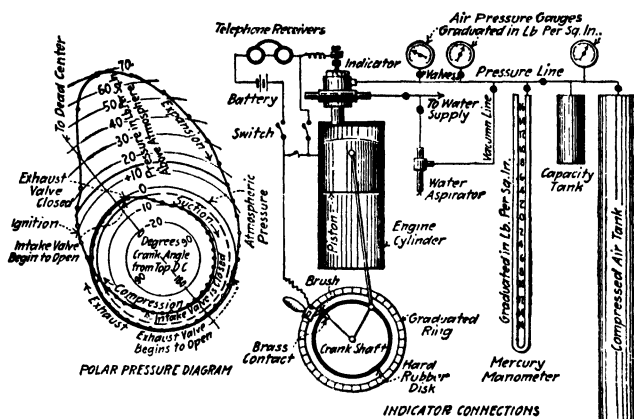


FIG. 107.— Details of Newell indicator.

Midgley Airplane-engine Indicator.¹—The optical type of engine indicator shown in Fig. 108 consists of the pressure element *A* and the camera *B*. In this instrument, the pressure element, in this case a piston, is located in the wall of the cylinder *C*. This avoids the usual errors that may be due to inertia effects and pressure-time shifts caused by relatively long connecting parts. The spiral spring *S* is mechanically connected so that it resists the upward movement of the piston in the cylinder *C*. The motion of the piston, which is always very small, is transmitted by a light rod to a bell crank, through which the position of a small flat mirror *M* is changed about a horizontal axis. The source of light is a small incandescent electric lamp from which the light is admitted to the camera *B* through a small circular opening *O*. Beams of light as reflected from the flat pressure mirror *M* fall upon an octagonal mirror *D*, which is either oscillated or revolved (depending on whether pressure-volume or pressure-time diagrams are desired), and is reflected from the octagonal mirror to a curved glass plate *E*. In the usual

¹ The *Midgley* optical indicator is described in detail in Diederich and Andrae, "Experimental Mechanical Engineering," p. 291.

operation of the instrument, indicator diagrams are traced on the curved ground-glass plate *E* where they can be seen by the eye. When photographs are desired, a curved clear-glass plate replaces the curved ground-glass and a sheet of photographically sensitive paper is fastened to the clear glass.

Calibration of Indicator Springs.—The pistons of engine indicators are invariably made of a very definite area, usually $\frac{1}{2}$ square inch; and it is possible to calibrate the deflection of the springs with respect to this area, so that a certain definite pressure per square inch¹ in the

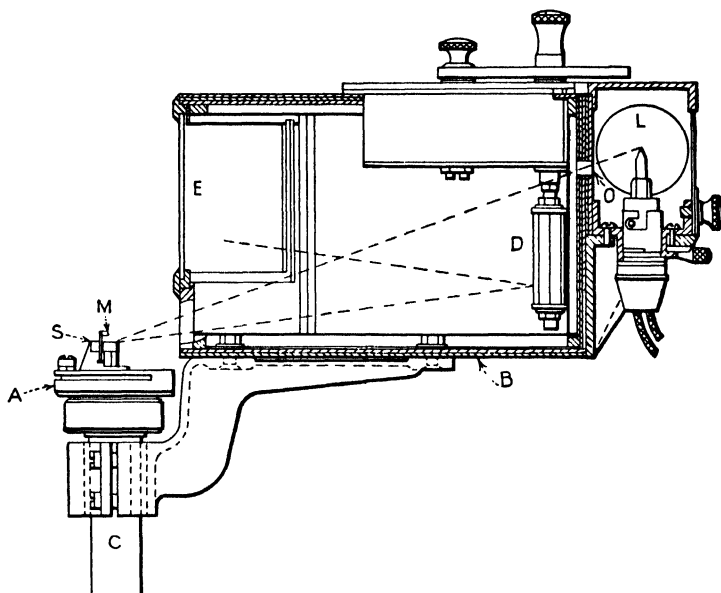


FIG. 108.—Midgley airplane-engine indicator.

cylinder will correspond to a definite deformation of the springs. In English units, the pressure on the piston in pounds per square inch corresponding to a vertical movement of the indicator pencil on the diagram of 1 inch is called the **scale² of the spring**. Indicator springs should always be calibrated by the makers. The calibration should be made when they are in the indicator in which they are to be used.

Apparatus for the Calibration of Engine Indicator Springs.—An apparatus similar to the one designed by Prof. M. E. Cooley is very generally used for the calibration of indicator springs. One of the latest and more elaborate forms of this instrument is shown in Fig. 109. In its essential parts this apparatus consists of a small cylinder *C*, sup-

¹ Indicators are always designed to relieve the pressure above the piston due to leakage around it, so that on this side there is always atmospheric pressure.

² Instead of "scale" the word "number" is often used. That is, a spring of which the scale is 40 pounds per square inch would be called "No. 40."

ported on a bracket *B*, a connection *I* at the top of this cylinder for the attachment of the indicator to be tested, and a stuffing box or gland at the bottom of the cylinder into which a plunger piston *P* is fitted. The lower end of this plunger rests on a sensitive platform scales. Any pressure in the cylinder *C* can therefore be weighed. Steam may be admitted to the cylinder through a pipe *E* and is exhausted through the pipe *A*. By adjusting the globe valves on the pipes *A* and *E*, any pressure desired can be secured in the cylinder *C*, and this same pressure is, of course, exerted both on the piston of the indicator above and on the

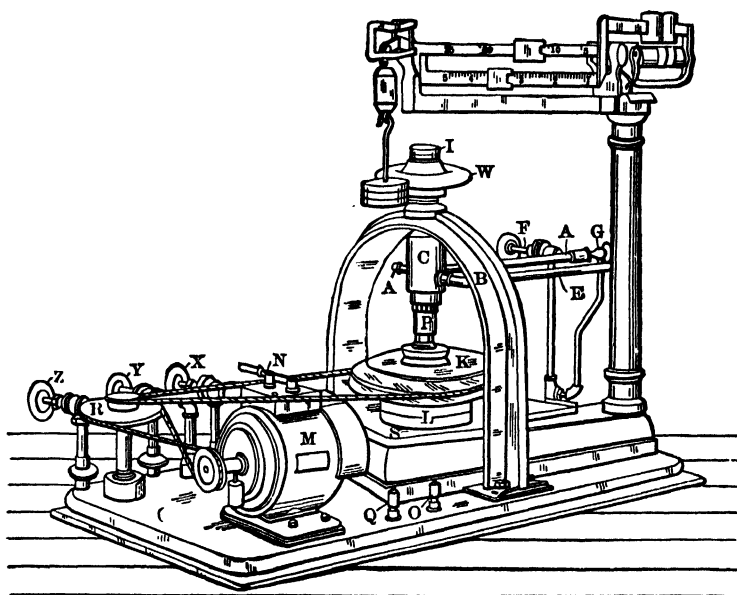


FIG. 109.—Cooley apparatus for calibrating indicator springs.

plunger *P* below. This plunger is usually made with an area of $\frac{1}{2}$ square inch. For a plunger of this area, then, if for a given pressure the scales balance at 10 pounds, the pressure in the cylinder *C*, and on the piston of the indicator, is 20 pounds per square inch. To eliminate friction¹ as much as possible, the plunger *P* should be kept spinning when observations are being taken. For this purpose a hand wheel *K* with considerable mass, for its "flywheel" effect, is provided on the shaft of the plunger. A more uniform motion of the plunger is obtained, however, by having the hand wheel grooved to take a small belt to be driven by an electric motor *M*. The plunger is supported usually on a ball-bearing joint set in a low pedestal *L*.

¹ The operation of the apparatus is usually much improved by pouring, just before the steam valves are opened, a few drops of cylinder oil into the cylinder *C* through *I* to lubricate the plunger.

By connecting the pipe *E* to a suitable manifold or similar fitting, to which are attached three separate pipes supplying, respectively, steam, air, and water under pressure, an indicator can be tested with varying pressures under the actual conditions in service, that is, when used for steam, air, or water. American engineers always prefer making calibrations of indicators under conditions as nearly as possible like those in ordinary use. The Power Test Committee of the A.S.M.E. recommends the following procedure:

To bring the conditions approximately at least to those of the working indicator, the steam should be admitted to the indicator in as short a time as practicable for each of the pressures tried, and then the indicator cock should be closed and the steam exhausted before another pressure is tried. By this means the parts are heated and cooled as under working conditions. For each required pressure open and close the indicator cock a number of times in quick succession, then quickly draw the line for the desired record, observing at the same instant the "reading" of the standard used for comparison. A corresponding atmospheric line is to be taken immediately **after** each pressure line.

Indicator springs for gas and oil engines should also be calibrated with the indicator in as nearly the same condition as to temperature as exists when it is in use. A simple way of heating recommended is to subject it to steam pressure just before calibration. Compressed air is a suitable fluid for the actual calibration, being preferred to steam, as it brings the conditions as nearly as possible to those of practice when the indicator is in actual use in gas or oil engines.

In Europe an apparatus like Fig. 110 is used a great deal. It is essentially the same as the dead-weight gage testers described on page 22, except that there is a connection for an indicator. Pressure is applied by loading weights on the platform *P* resting on the plunger. In this apparatus the gage *G* serves merely as a means of checking and avoiding mistakes. American engineers object to this method because the calibration is made when the indicator is under nothing like the conditions of service, at least as regards temperature. Many engineers calibrate their indicators by comparing them with a good test gage which has been carefully calibrated with a dead-weight tester. The gage and the indicator are put on the same pipe carrying high-pressure steam. The movement of the pencil of the indicator is carefully observed and compared with the reading of the gage. This latter is the method suggested by the Power Test Committee of the A.S.M.E. in their report in November, 1912.

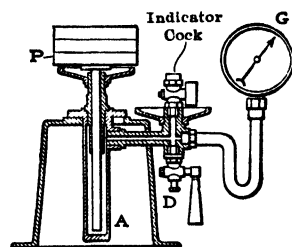


FIG. 110.—Dead-weight tester for indicator springs.

A somewhat simpler form of apparatus intended for the so-called "dry method" of testing is shown in Fig. 111. A suitable fitting for receiving the indicator *I* is supported on the bracket *B*. The legs of

this bracket span over a sensitive platform scales *S*. A small rod *R* rests at its lower end on a small pedestal standing on the platform of the scales *S*. On the top of this rod there is a cap supported on a small conical bearing to give some flexibility. This cap is made to fit easily into the lower side of the piston in the indicator. The indicator itself is attached to the top of the hand wheel *W*. Then when the hand wheel

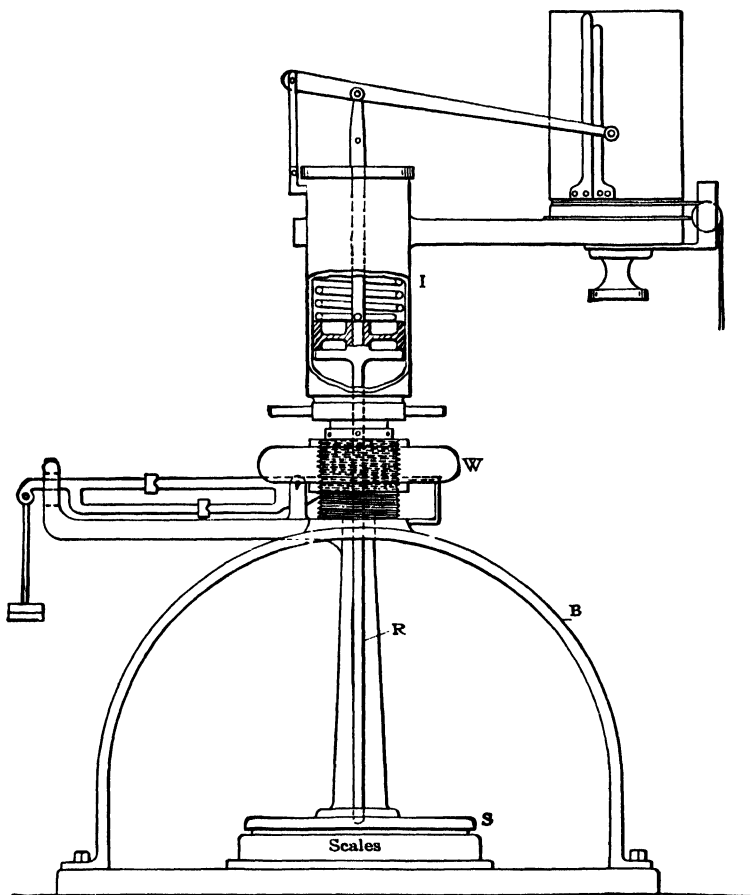


Fig. 111.—Apparatus for "dry" method of indicator spring testing.

is screwed downward, the indicator comes down with it and compresses the indicator spring. At the same time a pressure is exerted on the rod *R*, which can be balanced on the scale beam. When a force is applied to compress the spring in the indicator, the magnitude of the force can be determined by weighing the pressure on the scales. If the area of the piston in the indicator is $\frac{1}{2}$ square inch, then twice the weight on the scales is the pressure exerted in pounds per square inch. Heat

can be applied to the indicator by passing steam through a rubber or flexible copper tube wrapped around the cylinder.¹

Method for Calibration of Springs.—After the internal parts of the indicator are cleaned, the spring to be calibrated inserted, and the piston oiled with cylinder oil, the indicator is to be attached to the indicator cock on the calibrating apparatus. Before the card is put on the indicator drum on which the record is to be made, two approximately parallel and vertical lines should be drawn on it about $\frac{1}{2}$ inch apart, similar to the lines *AB* and *CD* in Fig. 112. Meanwhile the indicator should be

No. 1 Hour 1:30 P.M. 11/2/1919

Which End.....	A	C	Area.....
B. Press.....			Length.....
Vac. gauge.....			M. Ord.....
Revs.			M. E. P.....
Spring.....			I. H. P.....

Up ↓ Down

INDICATOR NO. 1620

OBSERVER: J. W. B.

"Atmospheric" Lines

FIG. 112.—Sample card illustrating a test of an indicator spring.

thoroughly warmed if a calibration with steam pressure is to be made. Then with the indicator cock and the valve on the steam pipe *E* (Fig. 109) closed and the exhaust pipe *A* open, draw the first calibration line on the card. This should be made by setting the pencil point at *D*, and then by pulling the cord attached to the drum draw a line **crossing** the vertical line *AB*. With springs of which the scale is 40 pounds per square inch or less, a similar record should be made for increments of every 5 pounds change in pressure, while for higher scales the increments may be made 10 pounds. If with increasing pressures the lines are drawn toward the left, then with decreasing pressures they should be drawn toward the right with equal increments, beginning at the opposite vertical line *AB*. By this method the corresponding lines for equal pressures will be imme-

¹ Some engineers advocate that an indicator should be slightly jarred just before each calibration line is drawn, intending that this jarring is equivalent to the vibrations which an indicator receives when in service on an engine. Since, however, readings are taken with both increasing and decreasing pressures, it is doubtful whether this additional work is necessary on an apparatus like Fig. 109. If, on the other hand, a dead-weight tester as in Fig. 110 or the method of comparison with a test gage (p. 21) is used, tapping both the indicator and the gage is probably very advisable.

diately over each other between the two verticals. With an accurate scale, graduated preferably to 0.01 inch, measure between *AB* and *CD* the distance from the atmospheric line first drawn to the various "pressure lines" and record the results. Care should be taken that with the increasing increments the pencil rises to the required pressure and that with decreasing increments it falls to these pressures. In other words, if, when the lines for increasing pressures are being drawn, the pressure

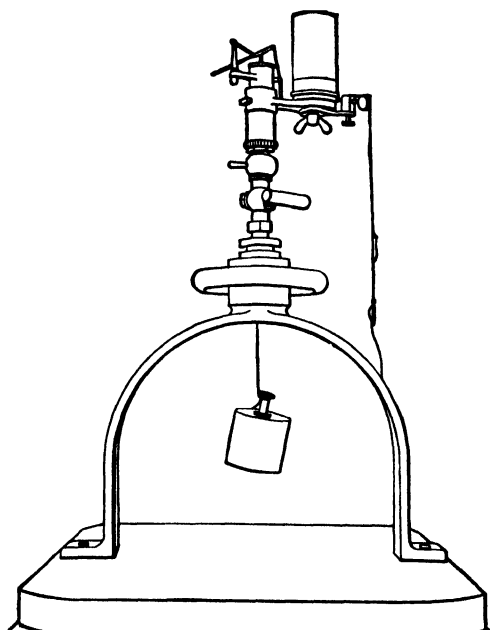


FIG. 113. —Apparatus for testing indicator springs in tension.

decreasing pressures shows the amount of friction and lost motion in the indicator.² The error of the instrument is obtained by comparing the mean ordinates of the card thus obtained with the actual pressures as determined by weighing. From time to time the accuracy of the platform scales should be determined by testing with standard weights. For dependable results two calibrations, each "up and down," should be made for each spring and the results compared.

When indicators are used for pressures which are never less than atmospheric, the springs in indicators like Fig. 97 are in compression and

rises too rapidly to draw the line at the proper time when the scale beam is just balancing, then the pressure should be again reduced below the value required, so that the pencil will be again ascending when the line is drawn. Similarly for decreasing pressures, if the pressure gets too low, it must be increased and again brought down to the required value.¹ If the results obtained do not seem to be consistent, the difficulty is probably due to passing the required pressure so rapidly that the lines have not all been drawn at the proper time.

The difference between the lines for increasing and

¹ The same precaution must be observed in beginning the test. To be sure that the pencil and piston have not been falling instead of rising, the piston rod should be pushed down lightly before the atmospheric line is drawn.

² Half of this difference, to be more accurate, represents the friction and lost motion in any one position.

apparatus of the form described are satisfactory; but when such indicators are used on the low-pressure cylinders of engines, the springs are usually in **tension**. For this service a slightly different device must be used for calibration. A suitable apparatus is shown in Fig. 113.

The indicator is supported on a bracket similar to the one used in the apparatus shown in Fig. 111. A short steel rod (about No. 18, B. & S. gage) is attached to the lower side of the piston in the indicator by screwing into a hole tapped centrally. Now if weights are suspended from the end of this rod,¹ the spring can be calibrated in tension by drawing lines on a paper card placed on the drum in the same general way as when the spring was tested in compression.

Suggested forms for the arrangement of data for these calibrations are given below. Form A is the one commonly used, and the results obtained by the suggested method of calculation are in accord with the recommendations of the Power Test Committee of the A.S.M.E. For more accurate work, form B may be used.

FORM A.—CALIBRATION OF INDICATOR SPRING (COMPRESSION)

In indicator No.

Rated scale of spring.
 Diameter of piston of testing apparatus. inches
 Area of piston of testing apparatus. square inches
 Identification marks on spring.
 Observers { Date.

No. of reading	Net weight on scales, pounds	Actual pressure on piston, pounds per square inch	Ordinates or "heights" measured on card, inches			True scale of spring, ¹ (3) ÷ (6)	Remarks
			Up	Down	Average		
1	2	3	4	5	6		

¹ "The calibration of a spring should be made for at least five equidistant points. For ordinary work the arithmetical mean of the various results should be taken for the average scale." (Report of Power Test Committee of A.S.M.E., 1912.)

¹ The weight of this rod and wires or strings supporting the weights must be added to them to get the correct tension. If, however, only the true scale of the spring is desired, as is usually the case, the weight of these parts need not be considered, provided, of course, the atmospheric as well as the other lines are all drawn with these parts attached to the piston.

FORM B.—CALIBRATION OF INDICATOR SPRING (COMPRESSION)

Date..... Observers.....
 Maker of indicator..... Type of indicator.....
 Scale or "number" of spring..... Area of plunger or tester...square inches
 Method of calibration..... Laboratory No.....
 Barometer..... inches of mercury..... pounds per square inch

Dead load		Check gage pressure, pounds per square inch		Ordinates measured indicator card, inches				True ¹ mean scale or "number," pounds per square inch (column 2 ÷ column 7)	Percentage error [(column 2 + column 9) ÷ column 2]
Net weight on scales, pounds	Actual pressure on piston, pounds per square inch	Reading	Corrected reading	Up	Down	Mean	Friction and lost motion (column 6 - column 5)		
1	2	3	4	5	6	7	8	9	10

¹ Column 9 gives the really important results of the test. The values given here are to be used as the true scale of the spring for values of mean ordinates of indicator diagrams nearest to corresponding values in column 7. A curve showing successive ranges of the scale of the spring should be plotted similar to Fig. 31 with true scale plotted against mean values of the diagram ordinates (column 7).

Curves.—Results should be shown graphically for calibrations of indicator springs by plotting for abscissas the actual pressures in pounds per square inch and for ordinates the corresponding average height, in inches, above the atmospheric line. Use very large scales, as otherwise these curves are of little value.

Reducing Motions or Driving Rigs for Indicators.—In most engines the length of the stroke is very much longer than the greatest possible movement of the drum of the indicator. It is therefore necessary to provide some means, called a reducing motion, which produces shorter movement, but which at every instant corresponds exactly with that of the cross-head. If this correspondence is not secured, the length of the indicator diagram cannot be accurately reduced or calculated, and the timing of the events or so-called "points in the stroke" will not be correctly represented.

The most satisfactory reducing motion or driving rig for an indicator is some form of well-made **pantograph** (Fig. 114) with a driving cord of fine annealed wire or a linen cord with a stranded wire core. Although unstretchable cord is to be preferred, it is not always available and ordinarily satisfactory results are obtainable with good hemp and linen cords if properly stretched in the process of manufacture.

Reducing motion and the cord or wire connections to the indicator should be so perfect as to produce diagrams of equal length when the same indicator is attached to either end of a double-acting engine cylinder. Tests of the reducing motion should show also a proportionate reduction of the motion of the piston at every point of the stroke; that is, there should be a fixed ratio in every position between the actual pro-

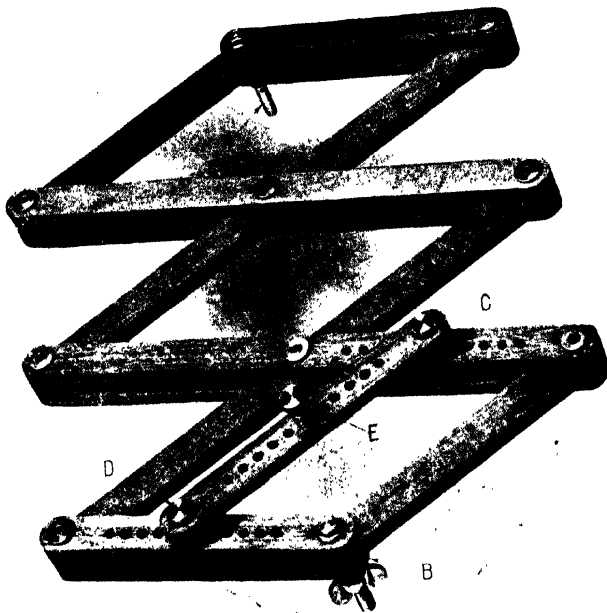


FIG. 114.—Pantograph or lazy-tongs reducing motion.

portion of the stroke passed through and the apparent proportion measured on the indicator diagram.¹

One of the commonest and most disastrous errors made in connecting up indicators is illustrated in Fig. 115, where an indicator cord is shown connected directly to the cross-head *C*. Obviously, near the position *C*₂ the movement of the drum is much less for a given piston displacement than at *C*₁. The difficulty is remedied in Fig. 116 by using a pulley *P* to avoid unequal angularity of the part of the cord connected to the

¹ To make this sort of test properly, the diagram should be taken just before the engine is stopped at a point other than a dead center, and this will serve as the basis of comparison with the measured length of stroke taken by the piston. The actual proportion of stroke moved through by the piston is measured by the distance the cross-head has moved compared with the full length of the stroke. Before making this measurement, the slack should be taken up by putting enough steam into the cylinder to bring some pressure upon the piston, but not sufficient to start the flywheel.

cross-head. Similarly, unequal angularity in a marked degree of the part of the cord **connected to the reducing motion** must be avoided for the same reason. The cord must leave the reducing motion as nearly as possible in a line parallel to the axis of the cylinder.

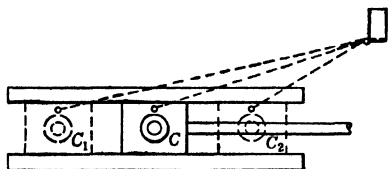


FIG. 115.—Incorrect method of connecting indicator directly to cross-head.

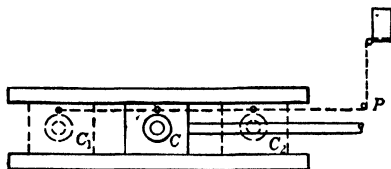


FIG. 116.—Correct method of connecting indicator to cross-head.

Pipes and cocks leading from the clearance space in the cylinder to the indicators should be as short and direct as possible. Except where no other device is practicable, the use of a three-way cock (Fig. 139,

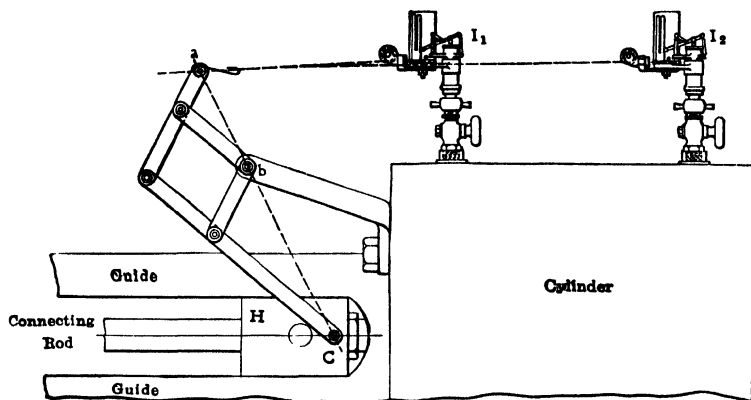


FIG. 117.—Simple pantograph for two indicators.

page 149) and a single indicator for a double-acting engine is not considered good practice. An indicator should be provided for each end for accurate work. The two indicators can be usually connected up to a single reducing motion in some such way as shown in Figs. 117 and 118.

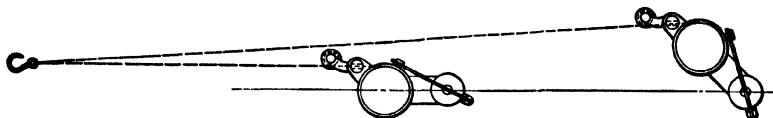


FIG. 118.—Approved method of connecting two indicators to one reducing motion.

Error produced by the use of three-way cocks is usually in increasing the area of the indicator diagram, owing to the tardiness of the indicator in responding to the changes of pressure.

One of the simplest forms of reducing motions is illustrated in Fig. 119. This device is pivoted at one point *A* to a pedestal supported on the frame of the engine and has a link *BH* connected to the cross-head. The indicator cord rides in a circular arc *CD*, proportioned to give the required movement to the drum of the indicator. Although this arrange-

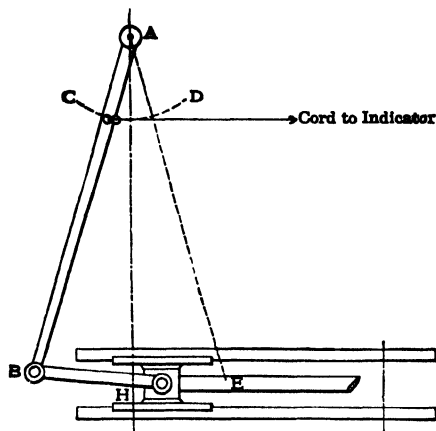


FIG. 119. — Single pendulum reducing motion

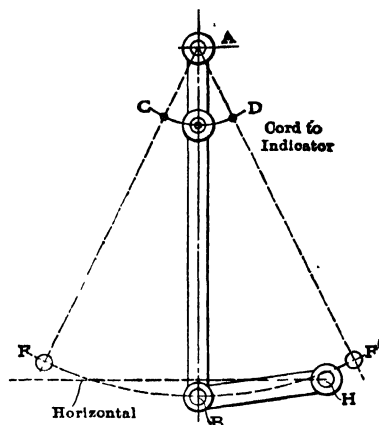


FIG. 120. — Approximately accurate pendulum device.

ment does not give an exact reproduction of the movement of the cross-head, yet if the pendulum *AB* and the cross-head are simultaneously at the middle of their strokes and the position *B* is as far below the horizontal drawn through *H* (Fig. 120) as the extreme positions *F* and *F'* are above it, the error is insignificant. An improved type of this device is shown in Fig. 121, in which the cord rides in a groove on the cir-

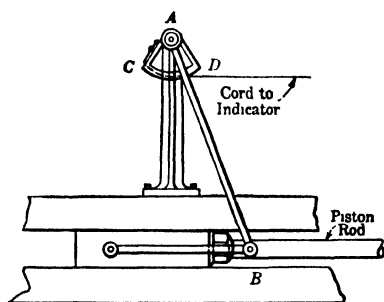


FIG. 121 — Maloney's reducing motion.

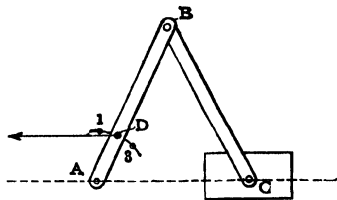


FIG. 122. — Oscillating-arm device.

cumference of a quadrant pulley. By attaching the pendulum to the quadrant pulley by means of a suitably designed "slip"-joint or clutch, the pendulum can be disconnected from the quadrant so that the segment and the indicator cord will be moved only when the indicator diagrams are to be taken.

A very good and simple device is shown in Fig. 122, consisting of two rods AB and BC , together with a pin D for the attachment of the indicator cord. If this is made so that $AB = BC$, then in any position of the arms the movement of the point D will be exactly proportional to that of the cross-head at C . The point A is fixed to the frame of the engine. The point B will always move half as far **horizontally** as C , because ABC

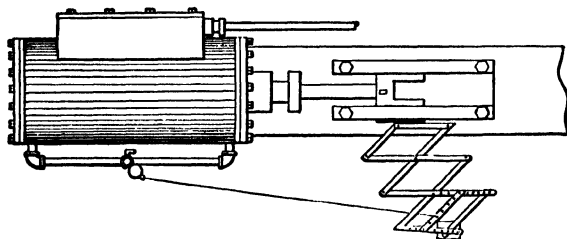


FIG. 123.—Plan view, showing attachment of pantograph.

is an isosceles triangle and the perpendicular drawn from B to the line AC bisects it. Then the **horizontal** movement of any point D in AB will be $\frac{AD}{AB}$ of that of B or $\frac{AD}{2AB}$ of the length of the stroke. Except for the very small change of the angularity of the cord as D goes from the position 1 to 3, the motion of the indicator drum moved by this device will be perfect.

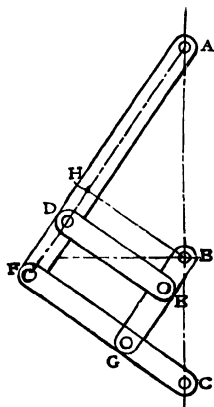


FIG. 124.—Simple parallel reducing motion.

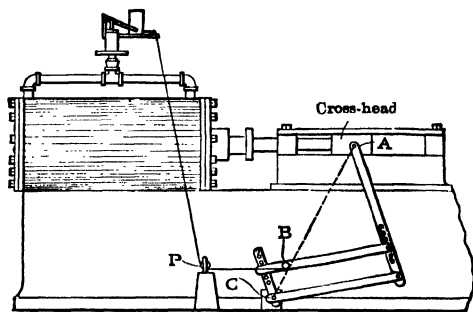


FIG. 125.—Simple parallel reducing motion as attached to a steam engine.

Figures 123, 124, and 125 are illustrations of modified forms of pantographs, sometimes known as **lazy-tongs**. Because of the numerous parts of which they are composed, requiring a great number of joints, they are likely to be troublesome with high-speed engines. A plan view showing one of the methods of attachment of this device to a hori-

zontal engine is given in Fig. 123. The obvious objections to the arrangement as shown are that there is more than the allowable angularity of the indicator cord between the pantograph and the pulley on the indicator, and that a single indicator is connected up by a three-way cock to both ends of a double-acting engine with a long stroke.

The ones illustrated in Figs. 124 and 125 are very commonly used. They are made usually of rods of iron or of steel nicely riveted together

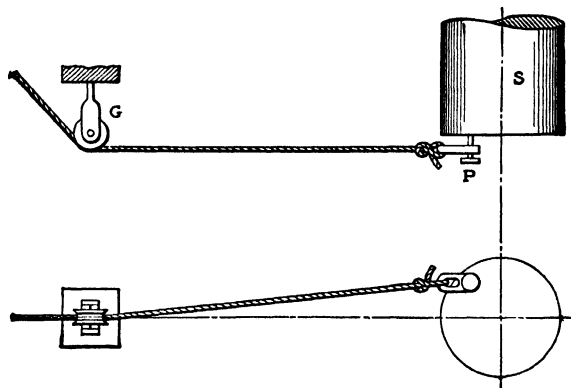


FIG. 126.—Inaccurate crank-shaft drive.

at the joints. The indicator cord is generally attached as at *B* (Fig. 124) and the ends *A* and *C* of the longer rods are fastened, respectively, to the cross-head and to the frame of the engine. It is a necessary requisite that in all these pantograph types the points corresponding to *A*, *B*, and *C* shall lie in a straight line as shown, and *DE* must be equal in length and parallel to *FG*. Then *AF* is in the same ratio to *HF* as the stroke of the piston is to the length of the indicator diagram.

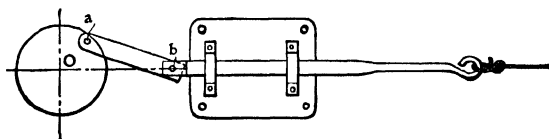


FIG. 127.—Crank and pin indicator drive.

It is often very convenient, especially in single-acting engines, to drive the indicator directly from the main crank shaft. A device of this kind often found in practice, but which does not give a true diagram, is shown in Fig. 126. The correct method to use in this case is illustrated in Fig. 127, applying a small crank mechanism. For absolute accuracy the ratio of the length of this connecting rod *ab* to the length *ao* of the "equivalent crank" should be the same as the ratio of the length of the connecting rod of the engine to the length of the engine crank.

The eccentric device (Fig. 128) is, of course, the same in principle as the crank mechanism. The only advantage for the latter is that it can be conveniently adjustable for a number of engines if the eccentric *E* is made in two halves with bushings suitable for adapting it to shafts of different diameters. The line of greatest eccentricity must, of course, be set exactly parallel to the engine crank (and in the same direction). This last statement applies also to the devices shown in Figs. 126 and 127.¹

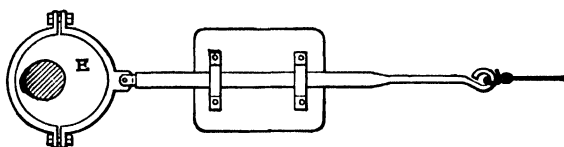


FIG. 128.—Eccentric type of indicator drive.

One of these crank-shaft devices must usually be applied in the case of single-acting engines, particularly where the connecting rod is attached directly to the piston. Another method often used on single-acting engines which do not have a closed crank case is a modification of that shown in Fig. 119. The stud *A* is supported in the usual way by a bracket bolted to the frame of the engine. A small fixture *F* (Fig. 129) provided

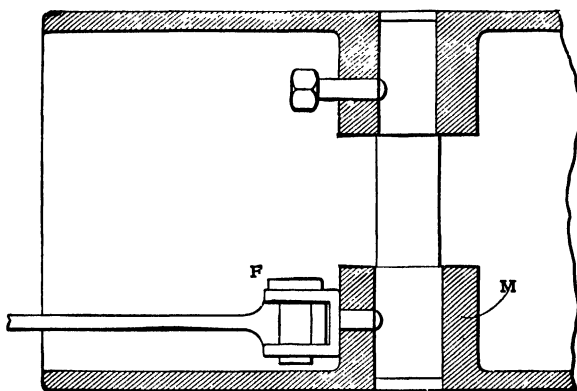


FIG. 129.—Device for single-acting engines.

with holes in the sides for the insertion of a small pin is attached, usually screwed, to the hub *M* of the piston. The swinging arm *BH* (Fig. 119) when pinned to the fixture *F* transmits freely the motion of the piston to the arm *AB*. Since both the point of attachment of the indicator cord and the point *B* move in arcs of circles about the same center *A*, the indicator drum will be moved in almost exact coincidence with the

¹ The error introduced by using the plain pin device is minimized by using a wire instead of a string up to the point of attachment of the hook on the cord attached to the drum of the indicator and making the length from the pin *P* (Fig. 126) to the guide pulley *G* as long as possible.

motion of the piston. The portion of the indicator cord attached to the arm *AB* should be moved as nearly as possible parallel to the line of the stroke of the engine for the best accuracy.

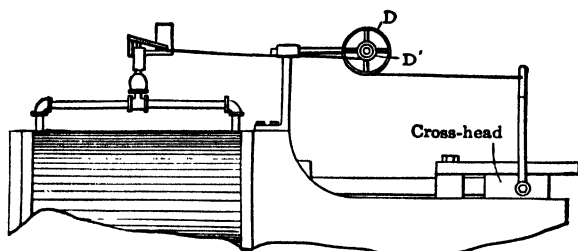


FIG. 130.—Reducing motion of concentric pulleys.

Reducing wheels which consist simply of a large and a small pulley attached to the same axis are coming into more or less general use. A typical arrangement is illustrated in Fig. 130. Pulleys *D* and *D'* are

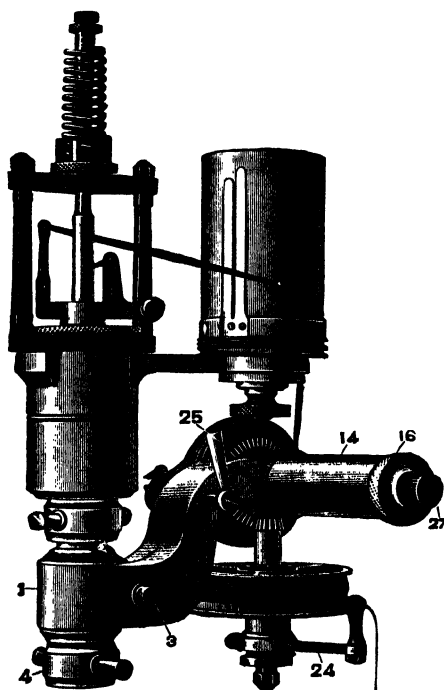


FIG. 131.—Indicator with reducing-motion attachment.

usually connected by a sliding sleeve so that they can be disconnected when indicator diagrams are not being taken.

Reducing wheels are not infrequently made for attachment directly to the indicator, as illustrated in Fig. 131. The figure shows the **Crosby**

reducing-wheel attachment. In Fig. 131 the cord on the wheel comes from the cross-head or from a bracket on it, and gearing drives the drum of the indicator. The ratio of the diameters of the wheels gives the reduction ratio. Each reducing wheel is provided with a "nest" of annular rings to be put on the small wheel to change its diameter, so that various reductions are easily arranged to suit the stroke of the engine.

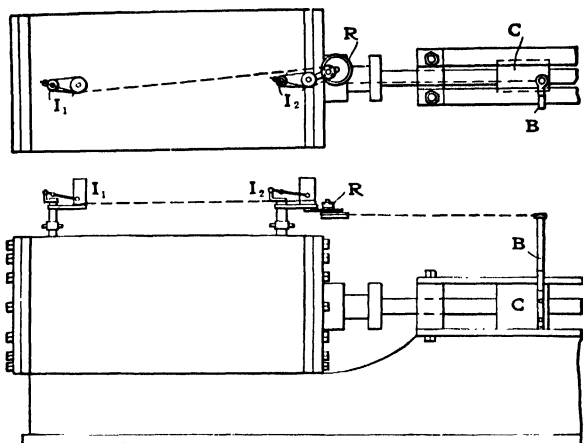


FIG. 132.—Single reducing wheel driving two indicators.

The device shown in Fig. 131 is started and stopped by turning the handle 14 and moving the clutch 25.

An arrangement of two indicators I_1 and I_2 to be driven from a single reducing wheel R is shown in Fig. 132. The connection to the cross-head C is by a bracket B . It is best in most cases to have a reducing wheel for each indicator in use, but this arrangement will often be very serviceable when, for example, one of the reducing wheels has broken down.

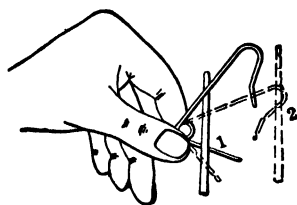


FIG. 133.—Trill's hook for indicator cord.

Some indicators are provided with **detents**, which are devices for starting and stopping the drum. For stopping the drum, they operate by engaging a pawl into teeth on the circumference of the drum near the bottom.

Obviously, the pawl must engage when the cord is pulled out to near the end of its stroke. After engaging the pawl, the cord will be likely to flap about on the return stroke, catch on something, and be broken on the next outward stroke. The best way to prevent this is to connect the string to a helical spring or rubber band to keep it taut.

In engine testing for long periods it is desirable to save wear of reducing wheels as much as possible by disconnecting the cord connecting

the pulley with the cross-head during the intervals between the taking of diagrams. Hooks like those shown in Figs. 133 and 134 are very convenient for this purpose, particularly if attachment can be made to pins or rods fastened to the cross-head. The hook shown in Fig. 133 is intended to be held between the thumb and finger at about an inch from either end of the stroke, so that the pin or rod on the cross-head strikes the straight part of the hook, and immediately the pin or rod will be caught in the hook as shown by the dotted lines. The one shown in Fig. 134 is also frequently used. Both hooks are intended to be pulled off the pin or rod when the cord is to be disconnected. Unless some special form of hook similar to the ones described is used, it is difficult to connect up the cord when a diagram is to be taken. Provision should also be made for preventing a disconnected indicator cord from being broken by getting tangled in the moving parts of the engine. Such difficulties can usually be avoided, with such a device as the one in



FIG. 134.—Simple hook for indicator cord.

Fig. 133, by continuing the cord from its point of attachment to the reducing motion closely past the indicator drum to a pipe or simple bracket that may serve as a stationary support. Between the indicator and this support a spiral or helical spring of light wire or a heavy rubber band is inserted in the cord. By this means the cord will always be kept taut and in motion when disconnected from the reducing wheel or from the indicator, as the case may be. If a ring is attached to the cord close to the reducing wheel and between it and the cross-head, it will be continually in motion, but it will be very easy to hitch into it the hook on the cord attached to the reducing motion. This method is particularly recommended in every case where wire is used instead of cord.

Relative Accuracy of Indicator Motions.—Pantograph types like Figs. 114, 117, and 123 to 125 give usually the best accuracy on small engines, if constructed accurately and used with short strings. The most important of the disadvantages is that invariably after a time they get shaky in one or more of the various joints, causing lost motion in the drive and many breakdowns in the indicator rigging. Pendulum types, like Figs. 119 to 121, are accurate enough for most work. The length of the pendulum should in either type be not less than one and a half times the length of the stroke.

Types using concentric pulleys, like Figs. 130 and 131, although suitable and quite serviceable for use on low-speed engines, are not accurate enough for good work on even moderately high-speed engines

and are always likely to make trouble on account of the indicator cords and wires getting twisted and torn. The inertia of the pulleys at high speeds tends to cause them to overrun at the ends of the stroke. Those having no means for mechanically starting and stopping the motion of the drum are also objectionable because of the difficulty usually experienced in hooking and unhooking the cord.

The crank-shaft drives illustrated by Figs. 127 and 128 are to be used only when one of the better methods is impracticable, as, for example, on single-acting engines. The device shown in Fig. 129 can be used also on some single-acting engines and is to be preferred to the others mentioned. It cannot be used on types of engines having a closed crank case like some small vertical steam engines, most automobile engines, and two-cycle gas engines.

All reducing motions should be designed to operate with a minimum number of guide pulleys.

Errors in Indicator Diagrams and in Calculations of Indicated Horsepower.—Taking indicator diagrams from steam or gas engines is regarded by most engineers as a very simple and commonplace operation, and consequently it is not unusual to find inexperienced persons placing great reliance upon indicator diagrams and the calculations therefrom, when, as a matter of fact, the conditions under which the diagrams were obtained make them in error possibly from 5 to 10 per cent. Except in the hands of an expert, the engine indicator, even of the best makes, cannot be relied on to give the mean effective pressure in the engine cylinder with a smaller error than 2 or 3 per cent. Principal sources of errors found in indicator practice will be discussed in the following sections.

1. Errors in the Reducing Motions.—The theory of the application of the engine indicator requires that the drum shall move in synchronism with the piston of the engine, following accurately its changing speed from one end of the stroke to the other. Errors in the application of this principle are usually found in the construction of the reducing motion. A familiar example of such errors is shown in Fig. 126 (see page 141), showing the drive for the drum of the indicator taken directly from a pin on the crank shaft of the engine. Another common error is in the application of parallel-motion drives like Figs. 124 and 125, when the points marked *A*, *B*, and *C* are not in line—in other words, so that a straight line will not pass through the three points. Stretch in the string, particularly if it is long, is a fruitful source of error. If the string stretches only a little so that it does not bring the drum up against one of the stops, this error is difficult to observe; but when the cord becomes so long that the diagrams drawn are not of the normal length, the effect can be detected by the knock of the drum against the stop and also by the distorted diagrams, of which Fig. 135 is an example.

The full lines show the kind of diagrams that are obtained when the indicator drum strikes one of the stops. The dotted lines show the parts of the diagrams that are missed.

2. Vibrations Due to Inertia.—Indicator diagrams often show irregular lines due to vibrations of the mechanism of the indicator, being particularly pronounced just after a sudden change of pressure like that occurring in a gas engine following the point of ignition. Figure 136 is an indicator diagram taken from a gas engine which shows such oscillations. The same sort of effect is observed in high-speed steam engines after the sudden changes of pressure that occur at the points of admission and of cut-off. The great fluctuation of pressure at admission causes the oscillation at *A* in Fig. 137, while the irregularities in the expansion line at *B* follow the closing of the steam ports at cut-off. Vibration

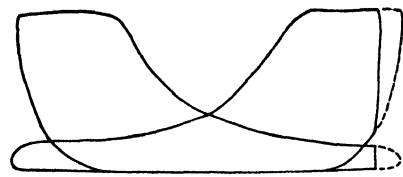


FIG. 135.—Result of having indicator cord too long.

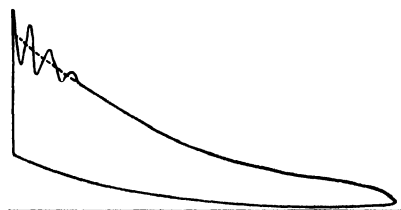


FIG. 136.—Oscillations in diagram from a gas engine.

effects can usually be best eliminated by using a stiffer or “stronger” spring.

It is well known that **frictional resistance will damp out vibrations**. For this reason some engineers advise pressing the indicator pencil rather heavily on the drum to eliminate vibrations. This method, however, is not recommended, as errors due to excessive friction may be introduced which are far greater than those that can possibly be due to the oscillations shown in the diagrams.

Practically only those errors resulting from the incorrect measurement of the areas of the diagrams will be attributable to vibrations. Generally, a mean line drawn through the oscillations so that the areas of the loops on one side of the mean will be equal to the areas on the other gives the average pressure. A mean line, shown dotted in Fig. 136, illustrates the method. Since it is usually difficult to follow accurately with a planimeter lines with long oscillations, it is often desirable to draw the mean line in all such diagrams and follow this with the tracing point of the planimeter.

In order to avoid inertia effects when taking indicator diagrams on high-speed engines, the makers of the Thompson indicator recommend the following scales:

Maximum pressure (gage), pounds per square inch.	6	30	60	80	100	150	200	250
Scale or "number" of spring.	12	24	40	50	60	100	150	175

For slow-speed Corliss engines the following scales may be used:

Maximum pressure (gage), pounds per square inch.	8	30	50	90	125	150	200
Scale or "number" of spring.	8	16	24	40	50	60	80

For engines operating at not over 200 revolutions per minute the Crosby Company recommends the selection of a spring making the height of the diagram for the maximum steam pressure at the engine about $1\frac{3}{4}$ inches. For high-speed engines it is desirable to take smaller diagrams to avoid excessive inertia effects of the indicator piston and pencil mechanism. It is also recommended that the length of the diagram should not exceed $2\frac{1}{2}$ to 3 inches.

3. Frictional Resistances.—That serious errors in indicator diagrams are often caused by excessive friction has already been pointed out, particularly as regards the friction of the indicator pencil on the paper. An

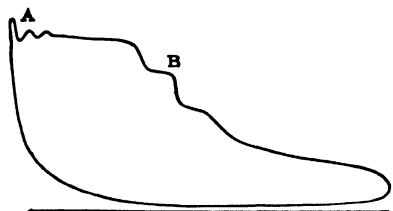


FIG. 137.—Oscillations in diagram from a steam engine.

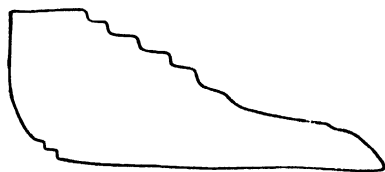


FIG. 138.—Diagram illustrating effect of a sticky piston.

example of excessive frictional resistance in the indicator cylinder is shown in Fig. 138. Friction in this case was probably due to a "sticky" piston, that is, one that did not move freely in the cylinder of the indicator. The peculiar "step-like" appearance of the expansion and compression lines is characteristic of this fault. Irregularities in the lines of diagrams due to friction cannot be corrected so accurately as can be done for pure oscillations, because in this case the pencil does not necessarily oscillate on the two sides of the true mean line equally but is almost continually **above** the true mean line of pressure in an expansion line, and correspondingly below when drawing the mean line of compression. In the case of distortion due entirely to pencil friction, the areas balance up fairly well, but tests show that the points in the cycle are late. It is therefore very important that the pressure of the pencil point on the paper should be carefully adjusted by the stop screw so that it makes a **fine, light, but clearly legible line**. With some types of indicators, specially treated¹ cards are provided on which a metallic

¹ An ordinarily good quality of paper can be prepared for use with brass points by applying to one side of the paper a thin coat of a mixture of about one part (by weight) of zinc oxide (ZnO), four parts of water, and one-tenth part of gum arabic.

pencil point is to be used. Diagrams taken on such cards show always very fine and clear lines, but it is doubtful whether such diagrams are as accurate as those taken with a good fine black lead pencil on a card of ordinary paper, since, except in the hands of the skilled operator, there is likely to be more friction in taking the diagram.

4. Errors Due to Faulty Indicator Connections to the Cylinder.—In many small power plants there is only one good engine indicator. On

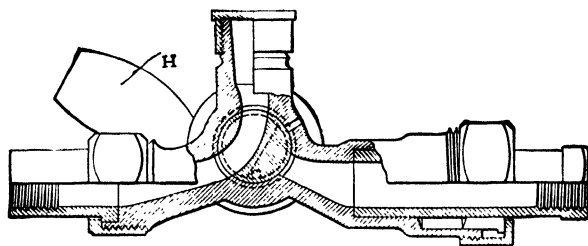


FIG. 139.—Typical three-way cock.

this account engines fitted only for commercial indicating are usually provided with a **three-way cock** (Fig. 139), which is used to connect either end of the cylinder with the indicator. This arrangement of piping is illustrated in Fig. 130 (page 143). For accurate testing, however, this sort of arrangement is not desirable, as the length of the pipe connections has an influence on the accuracy of the indicator diagrams. Long pipe connections have the effect of causing the indicator pressure to lag behind that in the cylinder¹ and may thus give a mean effective pressure 2 or 3 per cent too large, depending, of course, upon the diameter

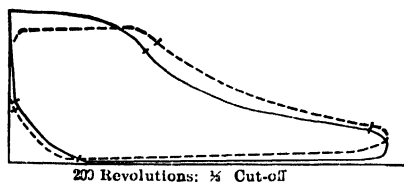


FIG. 140.—Diagram showing effect of long connecting pipes to indicators.

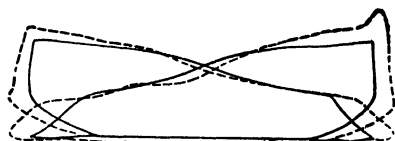


FIG. 141.—Diagram showing effect of stretching of indicator cord.

and length of the pipe connections. Pipes which are too small in diameter have very much the same effect as those that are too long. Figure 140 shows a distorted diagram caused by indicator connections that were much too long. The dotted line indicates the distorted diagram. A better arrangement is to screw an indicator cock directly into the end of each cylinder, and use two indicators, as shown in Fig. 117 (page 138).

The arrangement of a three-way cock and double-pipe connections should never be permitted on either ammonia or air compressors because

¹ Goss, W. F. M., *Trans. A.S.M.E.*, Vol. 18, 1896.

the clearance volume in compressors is usually reduced to the lowest limit mechanically permissible, and such pipe connections would increase the percentage of clearance enormously. In testing compressors and very small engines, the pistons of the indicators used should not leak much because leakage affects the degree of compression and leakages of ammonia fumes are most objectionable to those operating the plant.

5. Lost Motion in the Piston Connections and in the Joints of the Pencil Mechanism.—This is another fruitful source of errors. Fortunately, however, every engineer is usually able to get rid of lost motion due to looseness of joints without assistance.

6. Stretching of the Cord.—The effect on the diagram of the stretching of the cord when an engine was operating at 350 revolutions per minute is shown in Fig. 141. The full-line diagram was taken with a wire (no stretch) and the dotted line with a good quality of prepared hemp indicator cord. Stretching of the cord is due mostly to friction and inertia of the drum¹ of the indicator. It has been shown that the mean effective pressure is proportional to the stretching of the cord caused by drum friction and that the least amount of stretch in a good indicator cord is $\frac{1}{10}$ per cent of its length. In a cord 3 feet long, the stretching under the best conditions would be about $\frac{1}{10}$ ^{1.4} inch, making an error of about 5 per cent in the mean effective pressure in a diagram 3 inches long.

Fortunately, these errors due to inertia and friction have just opposite effects, the one making the diagram longer and the other making it shorter. The net result is apparently, according to reliable tests, that the indicated horsepower even at speeds of from 300 to 400 revolutions per minute is probably never in error more than 2 or 3 per cent if the indicator spring, drum spring, and quality of cord are properly selected for the pressure and speed, and it is assumed that the piston and drum move freely and are well lubricated.²

Diagrams obtained with the ordinary piston and pencil indicator are probably as accurate as other data in engine testing if the speed does not exceed 300 or 350 revolutions per minute, but at higher speeds such diagrams are regarded as of little value.³ Automobile and airplane engines usually operate at 2,000 revolutions per minute and over at maximum power, and special indicators are required (page 125). An instrument of this kind shows practically no inertia effects and no cord is used, to cause errors by its stretching.

7. Throttling.—Error in an indicator diagram due to not opening the indicator cock completely is shown in Fig. 142. Dotted lines are due to this throttling in the cock.

¹ REYNOLDS, OSBORNE, in *Proc. Inst. Civil Eng. (London)*, Vol. 83.

² *Proc. Inst. Mech. Eng.*, 1909, pp. 785-798.

³ Piston and pencil indicators are made by the II. Maihak Aktiengesellschaft, which give accurate diagrams at from 500 to 600 revolutions per minute.

8. Errors in the determination of the scale or "number" of springs are by far the most important in indicator practice. Power-plant engineers rely altogether too much on the calibrations made by the manufacturers of the indicator when it was new. That it is important to calibrate engine indicators with their springs frequently with some reliable and standard apparatus cannot be too strongly stated. It is absolutely essential that this work should be done both before and after every important engine test. There are not very many indicator springs which after considerable use check very closely to the scales for which they were intended. Errors in calculations of indicated horsepower are due more often to inaccurate springs than to any other cause. An engineer cannot safely assume that the true scale of the spring in his indicator corresponds at all accurately to the number stamped on it. It is not unusual to find springs in standard makes of indicators in error as much as 4 or 5 per cent.

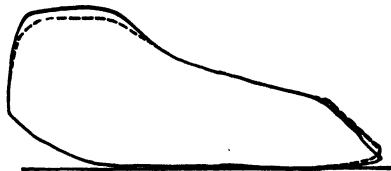


FIG. 142.—Inaccurate diagram due to indicator cock being partly closed.

Calculations of the indicated horsepower of an engine show usually the power developed on **one side** of the piston, which is commonly stated by the formula

$$\text{i.hp.} = \frac{p \text{lan}}{33,000} \quad (25)$$

where p = mean effective pressure on the piston, in pounds per square inch.

l = length of stroke, in feet.

a = net area of piston, in square inches.¹

n = number of revolutions per minute.

Of the terms of this equation only one, the mean effective pressure, is obtained from the indicator diagrams.

If we consider now only **one end** of the cylinder, the steam does work on the piston during a "forward" stroke, and, on the other hand, the **piston does work on the steam** on the "return" stroke. Hence to get the mean effective pressure for a stroke, the average pressure during the return stroke must be subtracted from the average pressure on the "forward" stroke; and this is obviously the same as the average length of all the ordinates intercepted between the upper and lower lines of the indicator diagram multiplied by the scale of the spring.

Usually the mean effective pressure is found by means of planimeters, the use of which for this purpose was explained on pages 100 to 115. An

¹ In all piston engines the area of the piston rod must be subtracted from the area of the piston on the side where the rod reduces the area effective for the action of the steam or other working substance.

engineer should, however, know how to calculate the mean effective pressure of an indicator diagram with reasonable accuracy without the use of such instruments. In such cases the **method of ordinates** is very convenient. With suitable draftsman's triangles¹ draw ordinates perpendicular to the atmospheric line at both ends of the diagram as shown in Fig. 143. Lay off on the edge *AB* of a piece of smooth flat paper a scale of 10 equal divisions so chosen that the total length of the 10 divisions is a little greater than the length of any of the indicator diagrams. This scale should then be placed **obliquely** across the diagram

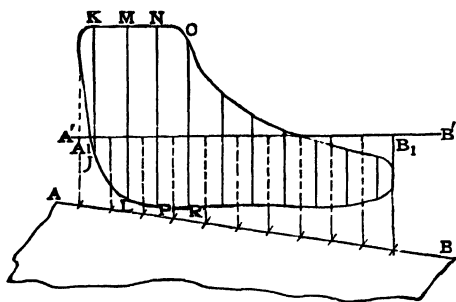


FIG. 143.—Diagram illustrating method of mean ordinates.

to be measured, so that the beginning and end of the scale will be located on the ordinates at the ends of the diagram. Now mark the diagram opposite the divisions of the scale with fine points, and at the **middle** of each of these divisions draw ordinates across the breadth of the diagram. The sum of the lengths of these ordinates divided by 10 gives the value of the **mean ordinate**,² and this when multiplied by the true scale of the spring gives the **mean effective pressure**. Some time can be saved in summing the ordinates if they are transferred with dividers one after the other to the edge of the strip of paper. The total length laid off divided by 10 is then the mean ordinate.

The Engine Constant for Indicated Horsepower.—In the use of Eq. (25) (page 151), where

$$\text{i.hp.} = \frac{plan}{33,000},$$

considerable time can usually be saved when calculating engine tests if the terms

$$\frac{la}{33,000} \quad (26)$$

called the **engine constant**, which always remains constant for **each end** of the cylinder, are first computed carefully and then used as constants throughout the calculations. In other words, the **indicated horsepower** is found for each end of the cylinder by taking the product of the terms,

$$\text{Engine constant} \times p \times n.$$

¹ Triangles to be used for this purpose should be tested for accuracy by setting on a straightedge and drawing a vertical line. Then turn over the triangle and observe whether the line drawn coincides with the edge of the triangle.

² Methods of calculating areas of irregular figures are given on p. 99. The area divided by the length gives the mean ordinate.

Horsepower of Automotive Engines.—The usual method of calculating the horsepower of automobile and airplane engines using gasoline for fuel is by the use of what is generally called the *S. A. E. formula* (Society of Automotive Engineers), although it has never been officially adopted by the Society. This formula is intended to determine only approximately the horsepower of a high-speed gasoline engine. Briefly,

$$\text{hp.} = \frac{\text{bore in inches} \times \text{bore in inches} \times \text{number cylinders}}{2\frac{1}{2}}$$

If D represents the diameter, or **bore**, of the engine cylinders in inches, and N represents the number of cylinders in the engine, this formula is stated as follows:

$$\text{b.hp.} = \frac{D^2 N}{2.5} = 0.4D^2 N.$$

For instance, if an automobile has six cylinders and the bore is 4 inches, the horsepower is,

$$\frac{4 \times 4 \times 6}{2.5} = 38.4 \text{ horsepower.}$$

This formula gives ratings much too low for modern high-speed automobile engines but gives with some degree of accuracy the maximum horsepower of four-cylinder engines which are not intended to run at more than about 2,000 revolutions per minute (piston speed 1,000 feet per minute and *mean effective pressure* about 90 pounds per square inch). Modern eight- and twelve-cylinder engines are intended to operate at piston speeds at least 50 per cent higher than the foregoing formula was intended to cover.

The Massachusetts Department of Public Works adopted in 1925 a different formula for horsepower ratings of automobile engines which is in better conformity with present automobile performance. In this formula the *length of the stroke* of the engine L in inches is one of the factors. If the other abbreviations as in the "S. A. E." formula are used, the *Massachusetts formula* is

$$\text{b.hp.} = \frac{D^2 NL}{10}.$$

This formula gives the same results as the one known as "S. A. E." when the length of the stroke is 4 inches.

Indicated Horsepower of Rotary Engines.—Figure 144 shows by a simple diagram a typical rotary engine. The steam inlet is at I and the exhaust is at E . A sliding blade P , corresponding to a piston, moves back and forth through the rotor R , as the latter revolves. In the figure,

P is shown at the point of cut-off, the dotted shading indicating the full charge of steam. Equation (25) can be written¹

$$\frac{pn}{33,000} \left[\text{volume swept through by piston} \left(\frac{\text{cubic inches}}{12} \right) \right].$$

A hole should be tapped through the casing near the top for the attachment of an indicator to determine the mean effective pressure p of the

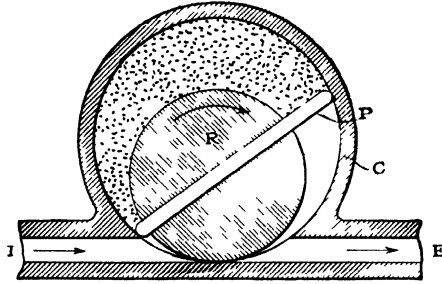


FIG. 141.—Diagram of typical rotary engine.

cycle. This hole can also be used to determine, by filling with water, the maximum volume in the cycle (general method explained on page 344), and these data, with the number of revolutions per minute n serve for calculating with considerable accuracy the indicated horsepower.

The steam consumption per indicated horsepower per hour for all types of such engines, if well made and when new, is about 100 to 125

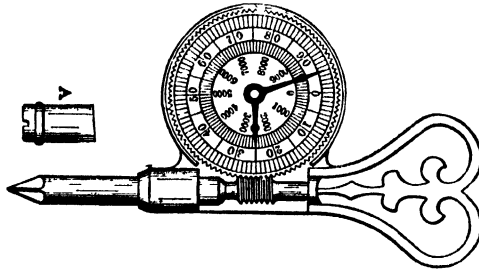


FIG. 145.—Starrett's differential-speed counter.

pounds. It is difficult to take up wear in such engines, so that after use for a short time much steam leaks through without doing work.

Speed Counters.—Some kind of mechanical counter is ordinarily used for determining the speed of engines and of other machinery with revolving shafts. For the usual services in testing, a hand speed counter

¹ In Eq. (25) l is in feet and a in square inches. The volume in cubic inches must therefore be divided by 12 to permit substitution in the equation.

(Fig. 145) is considered most reliable.¹ It is generally applicable, inexpensive, and accurate, so that every engineer should have one.

For slow-speed engines some type of fixed counter (Fig. 146) is frequently provided for attachment to the gage board in the engine room. For gas engines operating by a "hit and miss" method of governing, such fixed integrating counters are used in many places for counting the

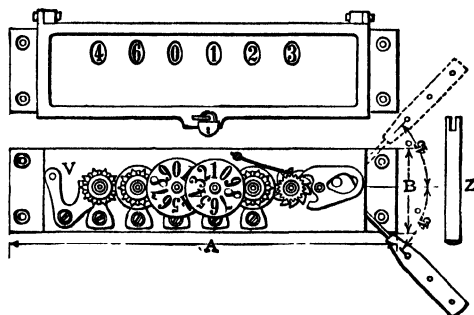


FIG. 146.—Integrating engine counter.

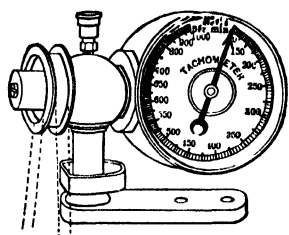


FIG. 147.—Belt-driven centrifugal tachometer.

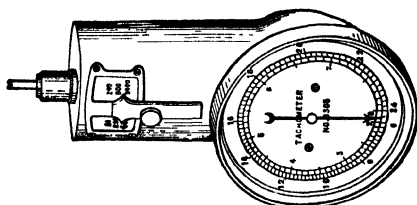


FIG. 148.—Hand type of centrifugal tachometer.

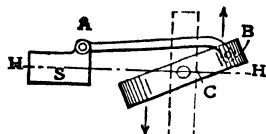


FIG. 149.—Sleeve and weights in centrifugal tachometer.

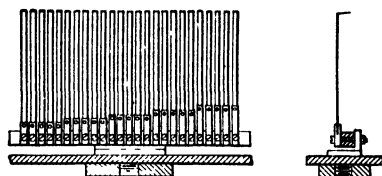


FIG. 150.—Details of vibrating-reed tachometer.

number of explosions. Actually the number of times the gas valve opens is counted. The greatest trouble with counters of this type is that they will sometimes "stick" even at the normal speeds of stationary engines.

Tachometers, operated centrifugally (Figs. 147, 148, and 149) or by the vibrating-reed method (Figs. 150 and 151), are not accurate enough for the determination of indicated or brake horsepower. They can be

¹ Starrett's hand counters are generally preferred both in America and abroad.

used conveniently, however, for observing roughly variations in the speed of steam turbines or electric generators when no accurate results are to be calculated from the observations. The vibrating-reed type operates by being placed on the frame of the machine, and the reed which is most nearly in synchronism with the vibration of the machine indicates by its excessive vibration the speed on a calibrated scale. Belted tachometers (Fig. 147) because of the added uncertainty regarding this slip of the belt are particularly unreliable.

Electromagnetic tachometers depend for their operation on the intensity of the magnetic drag due to flux generated which is proportional to the speed. They are simple in construction but rather delicate for commercial service and the temperature correction is usually difficult to compensate.

Fluid tachometers are essentially small centrifugal pumps discharging a colored liquid (usually alcohol colored red) into a vertical glass discharge

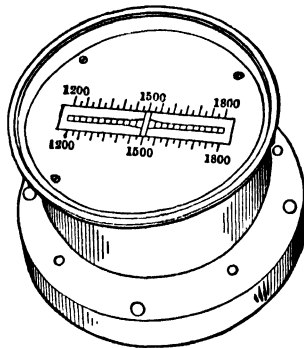


FIG. 151.—Commercial form of vibrating-reed tachometer.

pipe. The blades of the wheel are radial so that the instrument registers the same when running in either direction. The greater the speed, the higher the liquid will stand in the tube. Since the height to which the liquid will be forced in the tube varies approximately as the square of the speed of the wheel, the upper part of these tubes has a very much more open scale than near the bottom where accurate observations are difficult. Speeds less than 300 or 400 revolutions per minute cannot be satisfactorily observed with such instruments.

Combined Revolution Counter and Watch Movement.—The Hosler speed indicator shown in Fig. 152 is arranged to indicate the speed of an electric motor or the linear speed of a cutting tool. The indicator chiefly consists of two units: (1) a fine watch movement and (2) a geared revolution counter. These units are synchronized and the operation of the units is entirely automatic and requires no attention.

The instrument is to be held in the palm of the hand and the spindle is to be brought in contact with the shaft which causes the spindle to

revolve freely. The mechanism is set in motion by pressing the large button on the top. This automatically gears together the watch movement and the spindle. Both large and small pointers start to revolve and continue to do so for 3 seconds. At the end of this period the pointers are automatically disengaged and remain set, while the spindle continues to revolve. In reading the instrument it will be noted that the dial is calibrated with red and black figures. The red figures indicate

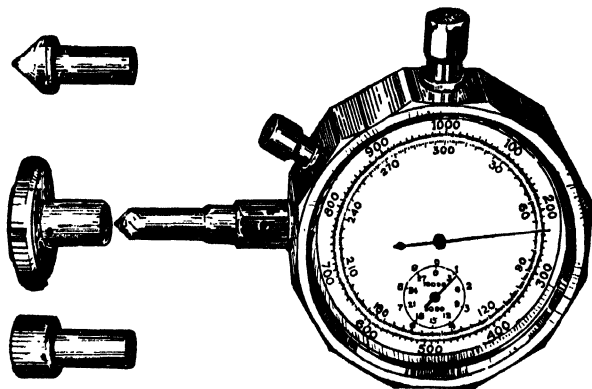


FIG. 152. -Combined revolution counter and watch.

the lineal speeds while the black figures indicate the revolutions per minute. Several consecutive tests can be made by continuing to press the button at the top of the instrument and the average of the readings will be the average revolutions per minute, or the lineal speed. In order to indicate the lineal speed, a small disk wheel is supplied with the instrument, which will slip on over the spindle. This disk wheel is to be brought in contact with the surface of the part to be indicated. This instrument as tested by the U. S. Bureau of Standards showed unusual accuracy.

CHAPTER VI

MEASUREMENT OF POWER—DYNAMOMETERS

A dynamometer, according to its derivation, is an instrument for measuring force or "power." These are of two kinds:

1. Those **absorbing** the power by friction and dissipating it as heat.
2. Those **transmitting** or passing on the power they measure, thus wasting only a small part in friction.

Absorption Dynamometers (Prony Brakes).—Of the class of dynamometers in which the power received is all **absorbed in friction**, the type generally used is called a **Prony brake**, named for Rev. John Prony, who many years ago developed a device of this kind for measuring power.¹ One of the simplest forms is shown in Fig. 153.

It consists of a lever *A*, from which a weight *w* is suspended from one end, and a block *B*, supported on a revolving drum or pulley *D*, is attached to the other end. A strap to which wooden cleats are fastened is held in place and tightened by the thumb nuts *N*, *N*. When the friction on the strap and block just **balances** the weight *w*, the lever arm *A* is horizontal and the apparatus is in adjustment. Stops, marked *SS*, are provided so as to limit the movement of the lever arm.

When the brake is adjusted or "balanced," the work done in a given time in producing the friction (the power absorbed) is measured by the weight moved multiplied by the distance it would pass through in that time if free to move. Then if

r = length of brake arm, in feet.²

n = revolutions of the shaft per minute.

w = weight on the brake arm, in pounds.

$$\text{b.hp.} = \frac{2\pi r n w}{33,000}. \quad (27)$$

In Eq. (27) the fraction $\frac{2\pi r}{33,000}$ is a constant quantity for a given brake and is called the **brake constant**. When a brake like the one in

¹ Strictly speaking, a brake of this kind does not provide means for directly measuring power, as, for example, horsepower, because the element of time is not indicated. In other words, it measures the **tangential force**, of which a couple (**torque**) in linear-weight units, such as foot-pounds, can be computed.

² The length of the brake arm is measured by the perpendicular distance from the line of action of the weight *w* to the center of the wheel. When the arm *A* is **horizontal**, as in Fig. 153, the length of the brake arm is usually measured by the horizontal distance from *P* to a line passing through the center of rotation perpendicular to the arm.

Fig. 153 is used, the **effective** weight of the brake itself as weighed at the point *P* must be **added** to the weight *w*.

According to Bach, suitable dimensions for a brake of this type are given by $bd = \frac{12 \text{ } b.hp.}{k}$ where *d* is the diameter of the brake pulley in inches, *b* is the breadth of the brake blocks in inches (usually about 1.5

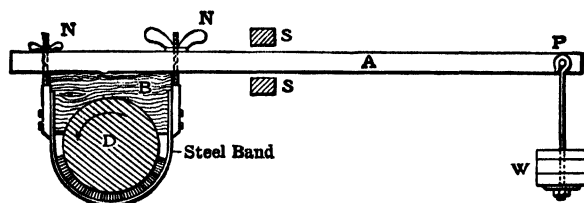


Fig. 153.—Simple Prony brake.

times the diameter of the shaft), *b.hp.* is the brake horsepower to be absorbed, *k* is 0.5 for air cooling and varies from 2.5 to 5 for water cooling as the speed increases.¹

A very common variation of the Prony brake is illustrated in Fig. 154. The block *B* in the preceding figure is replaced by a series of narrow cleats of maple or oak.

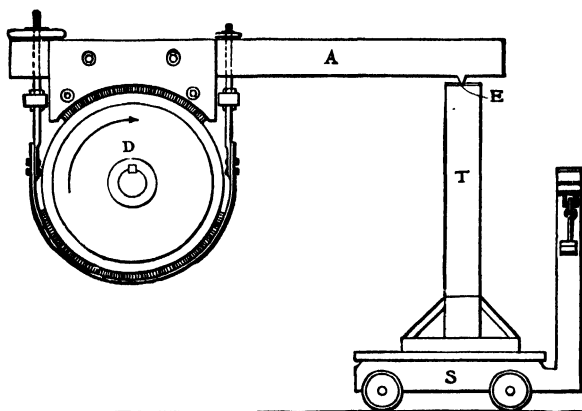


Fig. 154.—Prony brake with platform scales.

Rotation being in the opposite direction from that in Fig. 153, the knife edge at *E* on the arm *A* will now **press** on the pedestal *T*, and the weight *w* can be determined by weighing the pressure on a platform scales *S*. Since the scales receive not only the pressure due to the force producing friction but also that due to the weights of the brake and of the pedestal, these weights must be determined and are to be subtracted

¹ For information regarding the designing of Prony brakes for absorbing large powers, the reader is referred to "Engine and Boiler Trials, by R. H. Thurston, pp. 260-279.

from all of the readings of the scales to obtain the **net weight** w for substitution in Eq. (27). Weight of the brake and the pedestal, called the **zero reading**, must be obtained with the brake strap **slack**, so that the block B will rest as lightly as possible on the pulley. With small engines this zero reading is obtained most accurately by observing the weights on the scales when the brake pulley is turned around by hand first in one direction and then in the other. In this way we obtain for both the brake and pedestal, with rotation in one direction, the weight plus the friction due to their own weight and, with rotation in the other direction, the same weight minus the same friction. Half the sum of the two readings is, therefore, the weight corresponding to the pressure on the scales due to gravity alone. With large engines it is sometimes difficult to turn them uniformly by hand, so that the zero reading must be obtained by some other method. In practice this is done usually

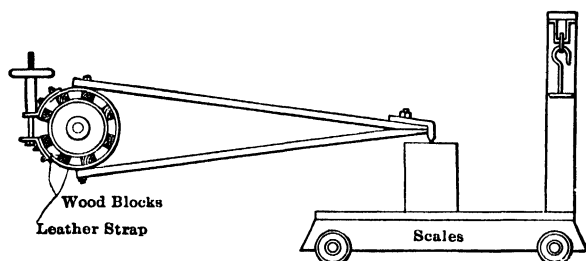


FIG. 155.—“Rough and ready” Prony brake.

by placing a very small rod at the top of the drum or pulley D (Figs. 153 and 154), vertically over the center of the shaft. Then if the strap is loose and due care is observed, the pressure on the scales can be obtained with sufficient accuracy without rotation. The cleats are attached to the bands by wood screws inserted from the outside and countersunk into the bands. Screws used for attaching the cleats to the upper block are inserted through the cleats and countersunk into the wood. At least $\frac{1}{4}$ -inch spaces should be left between the cleats to permit air circulation. In all such constructions for dynamometers, screws and nails should not touch the rubbing surface, as they are likely to cause the friction to be variable and the sound produced is objectionable. Many designers cut grooves into the inside surface of a few of the cleats. These grooves are to be filled with thick grease for lubrication.

A similar arrangement to Fig. 154¹ is shown in Fig. 155, showing maple cleats screwed to a leather belt. A piece of old belting is ordinarily used for this purpose so that by this method a very inexpensive Prony brake

¹ Although the type of Prony brake shown in Fig. 154 is very commonly used, the one shown in Fig. 155 is a better device as it has the point E on a level with the center of rotation of the shaft. As shown in Fig. 154, the tangential force at E has a component at right angles to the line of the standard T .

can be made quickly in any power plant. It has also the important advantage over the two preceding types that it is readily adjustable to different sizes of pulleys. Washers should be provided for the heads of the screws used to attach the cleats to the belt. If the inside surface of the pulley can be satisfactorily cooled by water, the rubbing surface of the cleats need be not more than 5 square inches per brake horsepower.

Another form of Prony brake is illustrated in Fig. 156, called a **strap brake**.

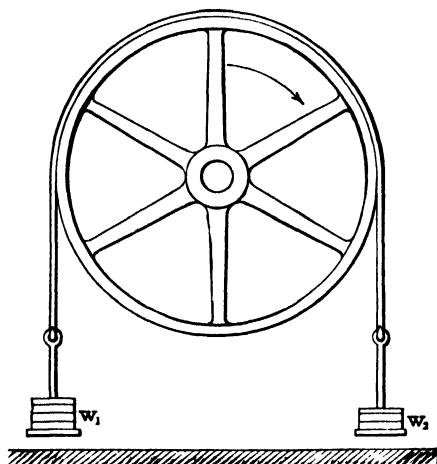


FIG. 156.—Strap brake.

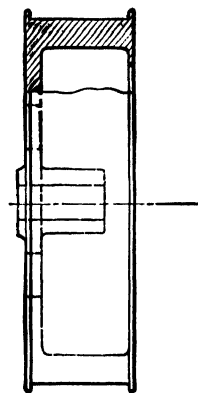


FIG. 157.—Pulley for Prony brake.

It is made up merely of a band of steel or leather or of strands of rope placed over or wrapped around a suitable pulley.¹ In this case weights must be suspended from both sides of the brake wheel or pulley.

In the case of the **strap brake** (Fig. 156) the net pull, corresponding to the weight w , in Eq. (27) (page 158), is $w_1 - w_2$. Now the same relation would hold if, as is often done, a spring balance is fastened to the floor on, say, the left-hand side; the pull registered by the spring balance

¹ It is desirable to use for Prony brakes pulleys of which the section of the face is a double "U," like Fig. 157. The outside rims are for keeping the brake in position on the pulley and those on the inside for receiving a small stream of water played upon the inside of the pulley. This stream of water by its evaporation will assist materially to dissipate the heat generated. Such brakes are often operated with pipes arranged to discharge water into the pulley and another pipe to carry it away. This is an excellent system, provided the latter pipe is used only to carry away a little overflow; but if so much water is used that there is practically no "steaming," the inside rim of the pulley will fill up with water to be spattered around in every direction as well as over the face of the pulley, where it is particularly objectionable, as it produces variable friction.

The cooling water should be admitted at the bottom of the wheel in the direction of rotation. The method is illustrated in Fig. 158 and shows the water-inlet pipe and also a flat-ended pipe, pointing in the direction opposite to the direction of flow of the water, for carrying away some of the water after it has traveled around the wheel.

would then be w_1 ,¹ and the net pull is, as before, $w_1 - w_2$. Brake horsepower is calculated then by Eq. (27), substituting for w the net pull $w_1 - w_2$, so that

$$\text{b.h.p.} = \frac{2\pi r n (w_1 - w_2)}{33,000}, \quad (28)$$

where r is the radius of the pulley plus half the thickness of the strap in feet, and n is the number of revolutions per minute. This type of strap brake is very accurate and sensitive but is suitable only for low

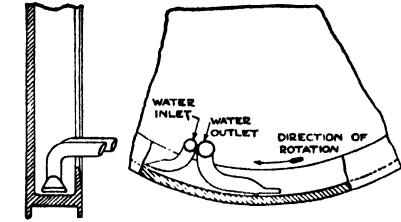


FIG. 158.—Water pipes for Prony brake pulley.

powers. About the same friction surface must be allowed as for wooden blocks in Bach's formula.

Rope brakes, like the one shown in Fig. 159, are much used for "commercial testing" of engines, as they are easily portable or can be made

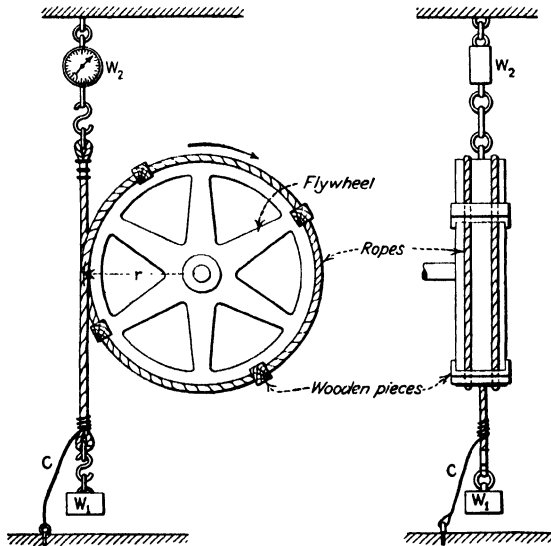


FIG. 159.—Rope brake with standard.

quickly at a small expense from materials always at hand. Moreover, they are *self-adjusting*, so that accurate fitting is not required. One of this type consists of a rope doubled around a pulley or flywheel on the shaft transmitting the power to be measured.

¹ To the pull w_1 must be **added**, however, the weight of any hooks placed between the end of the strap or rope and the spring balance; and if the balance is for any reason suspended in the **inverted** position, the weight of the balance itself must also be added.

As the friction at the rim of the pulley increases, the tendency will be to lift up the weight w_1 . The effect will be to reduce the tension in the end of the rope overhead connected to the spring balance and thus prevent a tendency to further increase the friction. Several U-shaped distance pieces of wood, preferably maple, are provided to prevent the rope from slipping off the pulley and to keep the parts of the rope separated. These distance pieces should be attached to the rope by soft iron or copper belt lacing, drawn in from the outside of the wooden pieces through the center of the rope, instead of being fastened with screws or nails on the inside, which will heat to a high temperature and then char the rope. Sometimes such fastenings when of hard metal will cut

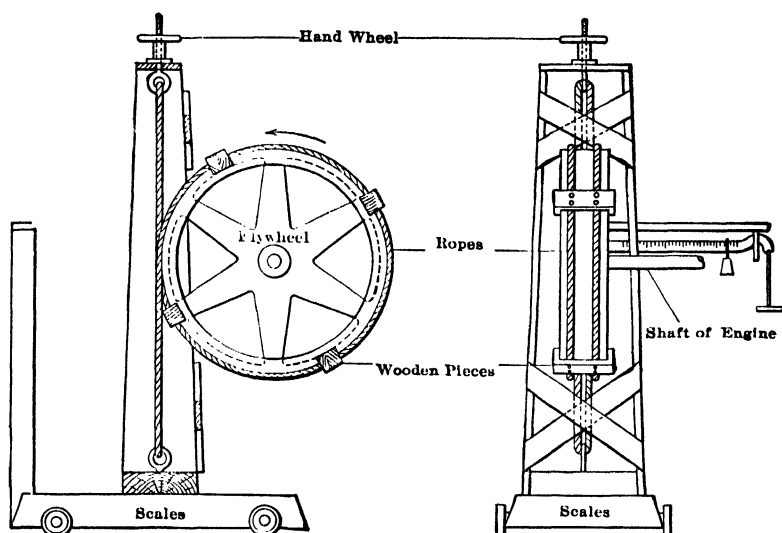


FIG. 160.—Rope brake with standard.

grooves into the surface of the pulley. In this arrangement the spring balance must be supported from some point overhead.

An **anchoring rope** c or safety stop securely attached to the weight w_1 should be provided to prevent the weights going over the wrong way when starting or stopping an engine, in case the valve is not very well set. Its weight or the weight of that part suspended from the weights must be included in w_1 . Similarly, the weight of the rope between the spring balance and the point where it touches the pulley should be deducted from the readings of the spring balance in accurate work. Spring balances require frequent and very careful calibrations. A modification of practically the same rope brake is shown in Fig. 160, where the rope is fastened at the top and bottom to a frame resting on a platform scales. This type of rope brake is not recommended where the pull on the tight side of the rope is more than 300 pounds.

Equation (28) is used also for calculating the brake horsepower for a rope brake, except that r becomes the radius of the pulley or brake wheel plus half the diameter of the rope, all in feet.

Rope brakes are best arranged with the rope placed on the pulley double as in the figures, so as to form a loop for supporting the weights. A brake made in this way of double $\frac{3}{4}$ -inch rope and provided with six cleats will absorb 50 horsepower if the pulley carrying the brake is about 4 feet in diameter, and the speed is about 300 revolutions per minute. For absorbing smaller powers half-inch rope with four cleats and a smaller brake pulley may be used. Manila or cotton rope is generally preferred. By steeping the rope in a mixture of deflocculated graphite and melted tallow its frictional properties are improved.

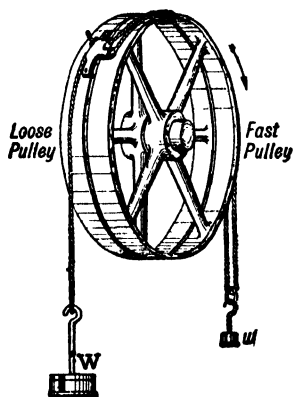


FIG. 161.—Prony brake of fast and loose pulleys.

The brake shown in Fig. 161 consists of fast and loose pulleys mounted side by side on the driving shaft. It is arranged that a rope clipped to the loose pulley hangs down over the fast pulley on one side and over the pulley to which it is fixed on the other. This is clearly shown in the figure. The frictional resistance between the fast pulley and the rope, when the pulley is revolving, causes the loose pulley to rotate through a small angle in the direction of rotation and in doing so it lifts a weight which is suspended by the rope hanging over it. At the same time, the rope over the fast pulley, which also supports a

weight, is by rotation of the loose pulley slightly unwound from the running pulley, thus reducing its arc of contact and, therefore, its braking effect. The resistance is thus automatically adjusted. The torque is, as in other cases, $r(W - w)$.

Water-jacketed Band Brake.—A unique band brake has been designed at the Pennsylvania State College, consisting of a flat strap with both sides made of copper. It is applied to a large flywheel of an engine, in the same manner as a rope brake. The ends are coupled together by an adjustable screw. The pressure of the brake strap on the wheel is produced and regulated by water flowing through the tube, much in the same way as in the Alden dynamometer (page 166). If the wheel surface is maintained in a well-lubricated condition, the wear of the copper tube is inappreciable. In a modified form of this brake a thin metallic band is interposed between the copper tube and the wheel, and consequently no wear of the tube takes place.

Fan Brakes or Dynamometers.—For determining the power of high-speed engines a dynamometer consisting of two flat fan blades arranged to be attached to the engine shaft is very convenient. Power is absorbed

may affect the power absorbed by a fan dynamometer as much as 20 per cent, that is, 10 per cent each way from normal.¹

Alden Brake.—An entirely different type of absorption dynamometer is the Alden brake, illustrated in Fig. 164.

In this apparatus the rubbing surfaces producing the friction necessary for absorbing the power are separated by a film of oil, and the heat generated is carried off by a stream of water circulating as indicated by arrows. It consists of a disk of cast iron *A*, which is connected to the shaft *S*, transmitting the power. This disk revolves between

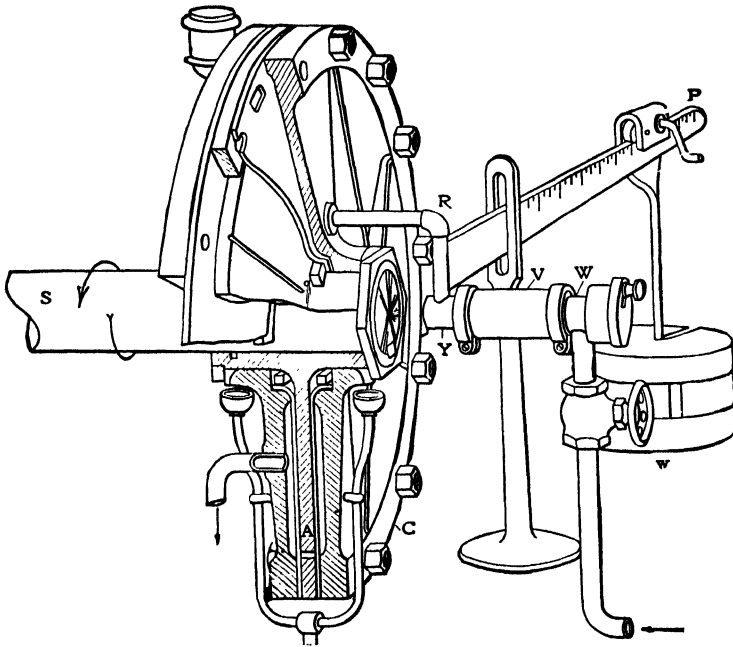


FIG. 164.—Alden brake.

two thin copper plates, fastened together at their outer edges to form a shallow cylinder which is filled with a bath of heavy (cylinder) oil. Water under pressure is discharged into the chamber adjoining the copper plates and any increase in pressure causes the copper plates to press toward the cast-iron plate *A* with more force. The friction of the thin film of oil between these copper and iron plates tends to turn those of copper, but, as they are rigidly connected to the outside casing *C* carrying the brake arm *P*, the tendency to turn can be determined by weighing as with a Prony brake. To maintain the moment of resistance constant under all circumstances, the pressure and consequently the flow of water into the casing is automatically regulated by a cylindrical valve

¹ See *Technical Notes*, National Advisory Committee for Aeronautics, No. 26.

V, which becomes partially closed if the brake arm moves above a certain horizontal position. This valve is shown in section in Fig. 165. The end at *W* is connected to the water main and the other end *Y* to the brake casing by means of a right-angled bend *R* (Fig. 164). Water entering by the pipe *W* passes through the ports *N* and then through the ports *H* into the pipe *Y*. Now a small angular movement of the pipe *W*, relative to the pipe *Y*, will open or close the ports *H* and thus regulate the supply of water. These ports are very narrow, so that a very small angular motion is sufficient to close them. By making the outer casing of the valve of rubber, it is found to be sufficiently flexible to permit

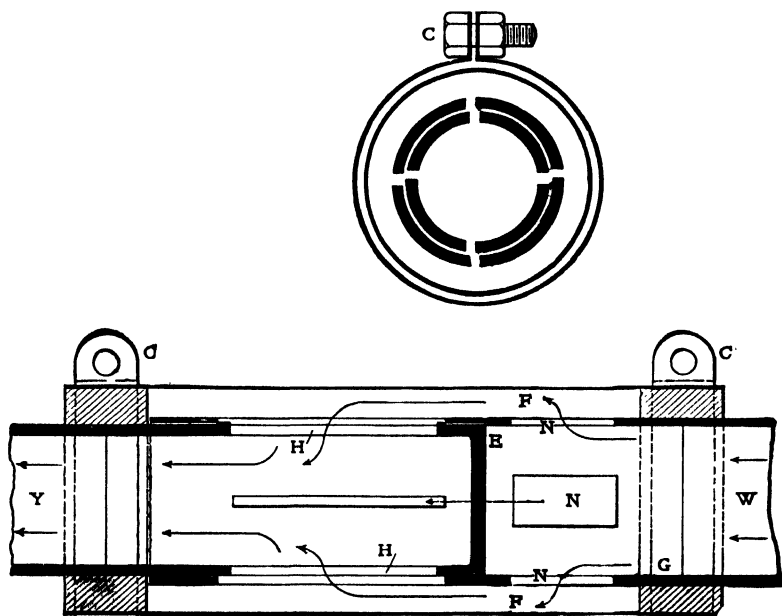


FIG. 165.—Regulating valve in Alden brake.

moving the valves and at the same time it offers very little resistance to the movement of the casing.

Reynolds-Froude Dynamometer.—Professor Osborne Reynolds has designed a modification of the Froude brake which is shown in section in Figs. 166 and 167. This device has a very large capacity for absorbing power. One of these brakes with a single rotor only 30 inches in diameter will absorb 700 horsepower at 200 revolutions per minute.

Water is supplied to the apparatus through the flexible pipe marked *E* (a rubber hose is very satisfactory), from which it passes in the direction of the arrows into the space *H* (Fig. 167), then into the centers of the vortices through the holes *J* (shown dotted) which are drilled through the walls of the pockets. From these pockets the water passes between

the rotor and the casing into the space *M*, from which it discharges through the discharge outlet *D* into the drain.

If any air collects in the center of the vortices, it can escape through the holes *I* in the pocket walls in the casing into the cored channel *L*

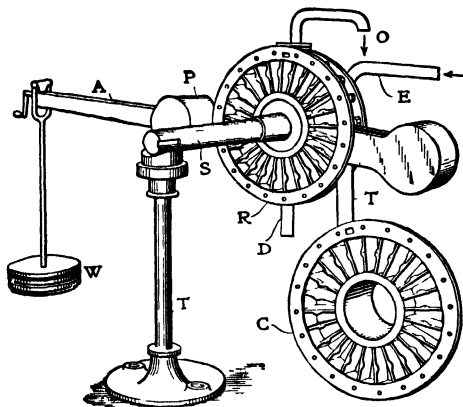


Fig. 166.—Reynolds-Froude dynamometer

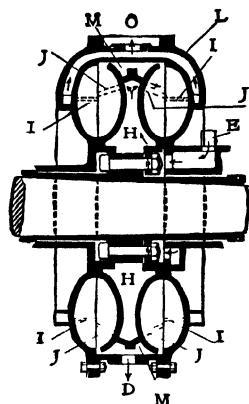


Fig. 167.—Reynolds-Froude dynamometer shown diagrammatically.

and the air outlet pipe *O*. If any water comes through these passages with the air, it is carried off through a funnel and pipe emptying into the main discharge pipe *D*. The brake horsepower (b.hp.) for this dynamometer is expressed on a very conservative rating by the equation

$$\text{b.hp.} = \frac{3.18}{10^{10}} n^2 d^5,$$

where *n* is the number of revolutions per minute and *d* is the diameter of the rotor in inches.

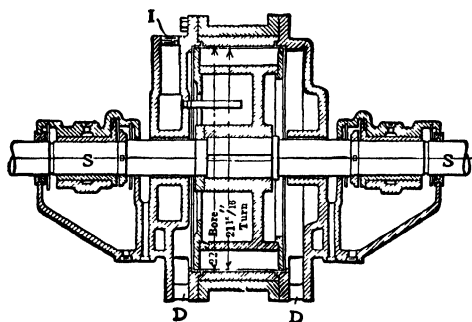


Fig. 168.—Simple water brake.

168 and 169 is also used for testing engines and turbines. The revolving paddle wheel or "runner" *R* is designed to run in very close clearance with the serrated rim piece *P*. The outer casing is rigidly attached to the lever arm *A* made of such a length as to facilitate rapid calculation. A little roller or wheel *W* on the end of this arm rests upon the platform scales used to weigh the load.

Water is admitted through a flexible hose connection at the opening *I* and from there enters the interior of the wheel. When the wheel is revolving, this water is thrown out by centrifugal force through small

Water Brakes.—Another type of brake shown in Figs.

holes drilled in the rim, entering into the outer tooth spaces. In this passage of the water a considerable fluid resistance is produced. The water finally escapes after passing through this tortuous passage through the close clearance around the outside of the wheel and discharges through the pipes connected at *D*.

Webb's "viscous" dynamometer¹ (Fig. 170) consists of a number of flat steel plates P_1 (circular saw blanks are excellent) fastened to a hub *H* which is keyed to the driving shaft *S*. Between these plates on the shaft are other

plates P_2 rigidly attached to the casing *C*. When a liquid is put into the casing, the rotating plates P_1 tend to carry it around with them by the action of viscosity, and the turning moment of the shaft *S* is communicated to the fixed plates P_2 and to the casing. The casing is supported on pedestals *E* provided with ball bearings. The turning

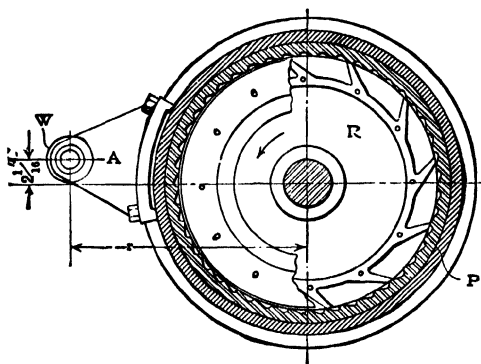


FIG. 169.—Section of brake shown in Fig. 168.

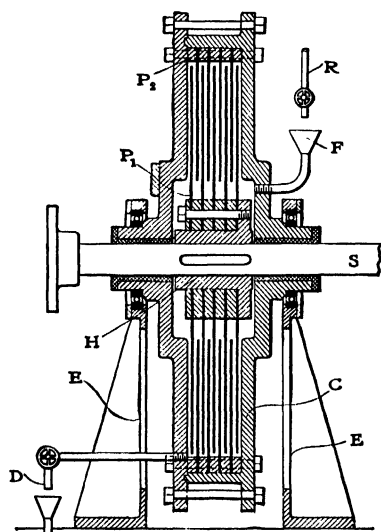


FIG. 170.—Webb's "viscous" dynamometer.

moment on the casing is balanced by weights placed on a horizontal arm as arranged for the Alden dynamometer (Fig. 164) or may be set up so as to press on a platform scale as in Fig. 155. The latter method is probably the better. Water is most commonly used as the liquid. It enters through the funnel *F* near the center of the casing and leaves at the discharge pipe *D*. A flow of liquid is maintained which carries away the heat generated. The power absorbed varies as the cube of the speed and the fifth power of the diameter. The frictional or "viscous" resistance can therefore be varied by adjusting the depth of the water in the casing. Usually a number of holes are made in both sets of plates through which water can

pass to the discharge connection without having the casing filled to the tips of the fixed disks before overflowing to the next compartment. The quantity of water in the casing can be regulated by both the inlet and

¹ *Trans. A.S.M.E.*, Vol. 24, p. 740.

discharge valves. It is generally best to adjust both valves at the same time for large variations in load. Viscosity of the water decreases somewhat with rise of temperature of the water.

Professor Webb has also arranged in some of his designs to supply the water through a hollow shaft and to regulate the supply by a piston valve to be adjusted axially to supply a varying number of compartments.

Regulation is accomplished not only by varying the radial depth of the water but also by changing the number of compartments containing water. In this latter arrangement no holes are required in the fixed or stationary plates to permit draining. Discharge connections are provided for each compartment so that there are as many drain pipes as compartments.

The dynamometer starts easily and without load so that it is better suited for use on steam turbines than those of the Alden and Froude types. The steel plates are the only parts subjected to high centrifugal stress and they are strong enough for all practicable speeds.

This apparatus is not large for the power absorbed. One of these dynamometers, provided with two disks, each 2 feet in diameter, absorbed 180 brake horsepower when running at 2,500 revolutions per minute with

a radial depth of 3 inches of water, and with 1 inch of radial depth of water 60 horsepower can be absorbed. Brake horsepower (b.hp.) absorbed by each rotating plate is approximately

$$\text{b.hp.} = \frac{n^3(r_2^5 - r_1^5)}{130,000,000},$$

where n is the revolutions per minute, r_2 is the radius of the moving plate in feet, and r_1 is the inner

radius of the annular ring of water in the casing.

Dynamos (Electric Generators and Motors) as Power Dynamometers.—One of the most convenient means for measuring the power of high-speed engines and turbines is to connect an electric generator to the main shaft as in Fig. 171. Then if the efficiency of the generator is known at the particular speed and output at which it is to be operated, a very accurate method of measuring the power of the engine or of any other type of motor becomes readily available. The output of the generator should be determined by observations of the volts and amperes with carefully calibrated *portable* instruments. Remembering that for direct-current generators volts times amperes gives watts and that 746 watts are equivalent to 1 horsepower, then if e.hp. is the horsepower output of the generator we have

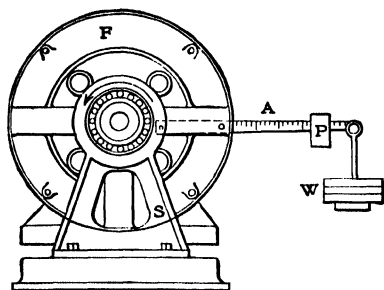


FIG. 171.—Electric generator used as power dynamometer.

$$\text{e.hp.} = \frac{\text{volts} \times \text{amperes}}{746}$$

It is not unusual to hear this result called the **electrical horsepower** of the engine or turbine.¹

The **actual horsepower delivered to the generator** is, of course, the **brake horsepower**, and into this result the efficiency of the generator enters. Thus, for direct- (or continuous-) current generators

$$\text{b.hp.} = \frac{\text{volts} \times \text{amperes}}{746 \times \text{efficiency of generator}}$$

The load on the generator should be maintained uniform by absorbing the electrical output in lamp or wire resistances for small powers, but for larger powers a water resistance or rheostat is generally used. The electrodes of a water rheostat are generally made of $\frac{1}{4}$ - to $\frac{1}{2}$ -inch iron or steel plates, allowing about 1 square inch per ampere.

As a rule, electric **motors** are very serviceable in mechanical engineering laboratories as power dynamometers. The efficiency is easily obtained, the usual method being to determine an efficiency curve for varying power inputs by a Prony brake test. This efficiency is

$$\text{Efficiency of motor} = \frac{\text{b.hp.}}{\text{e.hp.}}$$

where e.hp. is the "electrical"-horsepower input, as measured with voltmeters and ammeters.

If then a pump, air compressor, ventilating fan, or a similar machine is to be tested by the electrical method, it should be direct-connected to the shaft of the motor, and its efficiency E will be

$$E = \frac{\text{u.hp.}}{\text{e.hp.} \times \text{efficiency of motor}},$$

where u.hp. is the useful work done by the machine, in horsepower.

This last method serves also as a convenient method of obtaining the efficiency of generators, since by connecting a generator directly to the shaft of a motor previously "calibrated" (for efficiency) the electrical output of the generator and the input to the motor are readily determined.

When a so-called **variable-speed** motor is used as a dynamometer, its efficiency must be determined at the particular speed and power at which it will operate when driving the machine to be tested.

Eddy-current brakes² are built with a number of electromagnets and one or more copper disks. Either the coils or the copper disks may be

¹ When an alternating-current generator is used, the power factor must also be measured with a suitable instrument. In this case, the actual electrical horsepower is (volts $\times$ amperes $\times$ power factor) $\div$ 746.

² For more information regarding the design of this type see "Eddy-current Brakes" in *Jour. Inst. Elec. Eng. (London)*, Vol. 35, 1904-1905.

rotated with the shaft while the other is held stationary. Eddy currents generated by rotation with the electromagnets excited produce a resistance of which the moment is measured by a lever arm and scales as with similar forms of dynamometers.

Transmission Dynamometers.—Instruments of this type are used to measure the amount of power transmitted without absorbing any more power than is absolutely needed to move the dynamometer.

Goss Belt Dynamometer.—One of the simplest forms of transmission dynamometers, designed by Prof. W. F. M. Goss, is illustrated in Fig.

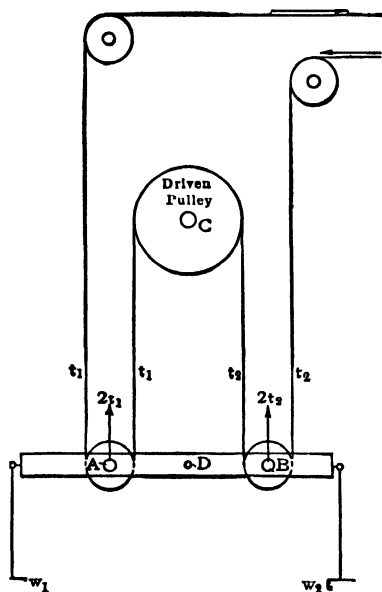


FIG. 172.—Goss belt dynamometer.

172 by a line drawing. Being so simple in construction, it can readily be made in any factory or workshop with the materials available. It is essentially a **differential lever** measuring the difference in tension between the two sides of a belt. This lever is pivoted at the point *D* and to it are attached the shafts carrying the pulleys *A* and *B*. Weight hangers are attached to the ends of the beam. The beam and the hangers must be balanced to be in the horizontal position, that is, in the position of equilibrium when the belt is not moving. Power transmitted is measured by the product of the speed of the belt and the difference in belt tension between the two sides of the dynamometer.

The force tending to raise the left-hand end of the lever is twice the tension t_1 of the tight side of the belt, while that raising the right-hand side is twice the tension t_2 of the slack side of the belt. These tensions on the two sides of the lever can be measured by weights w_1 and w_2 suspended from the hangers. The force tending to rotate the lever is therefore twice the difference in tension on the two sides of the belt $2(t_1 - t_2)$ and this force acts with a leverage $AD = DB = r$, when these two arms are made equal. If now the distance from the pivot *D* to the point of support of the weight w_1 is made twice $AD = 2r$, and if the weight w_1 is made equal to the difference in the tensions, it will balance the lever. For these conditions, taking moments about *D*,

$$\begin{aligned} w_1 \times 2r - 2t_1r + 2t_2r &= 0, \\ 2w_1 &= 2(t_1 - t_2), \\ w_1 &= t_1 - t_2. \end{aligned}$$

Now in any transmission system the difference in the tension of a belt or a rope on the two sides of a driven pulley multiplied by its speed is a measure of its power. If, then, d is the diameter of the driven pulley C , plus the thickness of the belt or rope in feet, n is the number of revolutions per minute of this pulley, and w_1 is the weight in pounds on the left-hand side required to **balance** the lever; when power is transmitted, then,

$$\text{hp. transmitted} = \frac{\pi d n w_1}{33,000}. \quad (29)$$

To reduce the vibrations of the apparatus, a dashpot is connected to the right-hand side. To prevent excessive movement of the lever when unbalanced, stops are placed above and below the lever on the left-hand side.

Another form much used for mill purposes and fan testing is the belt dynamometer, shown in Fig. 173. The driving pulley is A and the driven

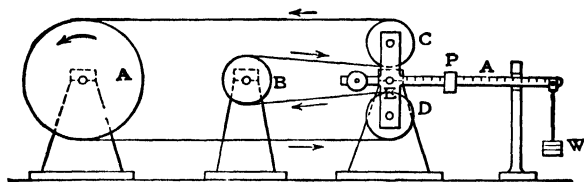


FIG. 173.—Compact belt dynamometer.

pulley is B . The connecting belt passes over the pulleys C and D . These two pulleys are mounted on the frame of the dynamometer carrying a scale beam A , all of which turn on the center of support. The difference in the total stress on the two sides of the belt is computed by multiplying the net weight on the beam by the equivalent leverage, the latter being found by dividing the length of the beam by the distance from the center of support to the center of pulley C or D .

Differential Dynamometers.—The apparatus illustrated in Fig. 174 is typical of a number of dynamometers indicating by means of a differential lever operated by gearing the amount of power transmitted.¹ This is a very common form of **transmission dynamometer**. Power is received from the motor (or engine) by the shaft A , which is connected only indirectly by means of gears to the shaft A' opposite, which transmits the power to the work. To the adjoining ends of these shafts bevel wheels B and D are attached. The lever L turns on an axis **concentric** with the shafts A and A' , in a plane perpendicular to them. It carries bevel wheels C and C_1 , gearing with B and D , through which the power is transmitted. There is a tendency then for the left-hand end of L to

¹ Similar forms of differential dynamometers are known as White's, King's, and Bachelder's. The first instrument of this kind, it is stated, was invented by Samuel White in 1780. These dynamometers are sometimes called **epicyclic**, signifying "wheels traveling around a circle or around another wheel."

go downward and for the right-hand end to rise. If, furthermore, the lever L were permitted to revolve, no work would be transmitted from B to D , and therefore D would remain stationary. As these gears are usually proportioned so that B revolves with half as many revolutions in a given time as L , a force applied at B at a given radius from the center will balance a weight twice as large at the same radius on the lever L . Then the moment of the force applied to the lever L to balance it must be twice as great as the moment of the force transmitted from either C or C_1 to D . That is, the sum of the moments applied to C and C_1 must

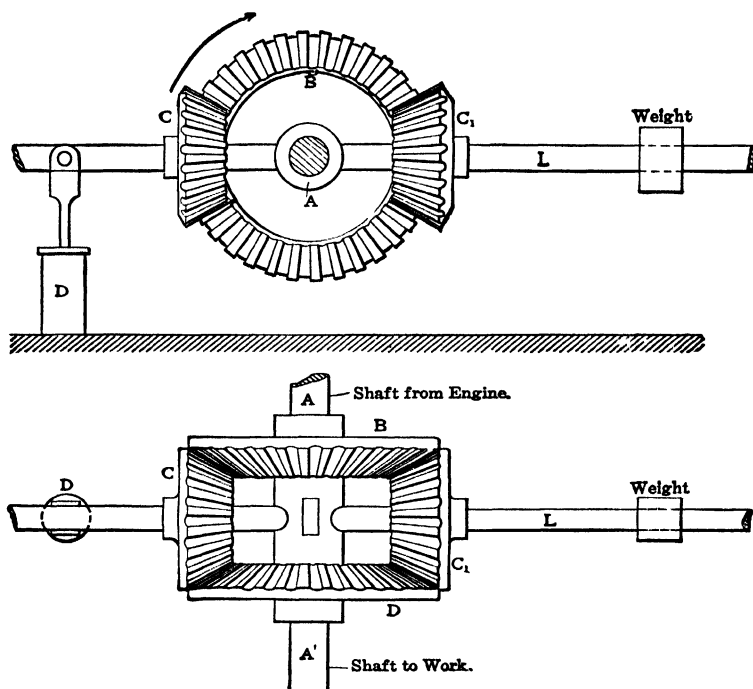


FIG. 174.-Typical differential-lever dynamometer.

equal the moment applied to B . If, therefore, l is the length, in feet, of the arm at which the weight w , in pounds, is applied, and n is the number of revolutions per minute of the shaft A , then the **power transmitted per minute** =

$$\frac{2\pi l n w}{2} \text{ foot-pounds,} \quad (30)$$

and

$$\text{hp.} = \frac{\pi l n w}{33,000}. \quad (31)$$

Now if the lever L is to be prevented from turning about its axis, a couple will be required which is twice the driving couple being trans-

mitted. If, then a weight of w pounds sliding on L as shown in the figure is placed at a distance l feet from the axis, so that the lever remains horizontal, the driving couple can be determined; and when the revolutions of the shaft A are known, the power can be found. A dashpot D is usually attached to the differential lever L to reduce vibrations. This dashpot should always be kept filled with glycerine or good clean oil. If the dashpot is sticky, consistent results cannot be expected.

A Webber differential-transmission dynamometer as made commercially is illustrated in Fig. 175. The scale on the lever arm of this instru-

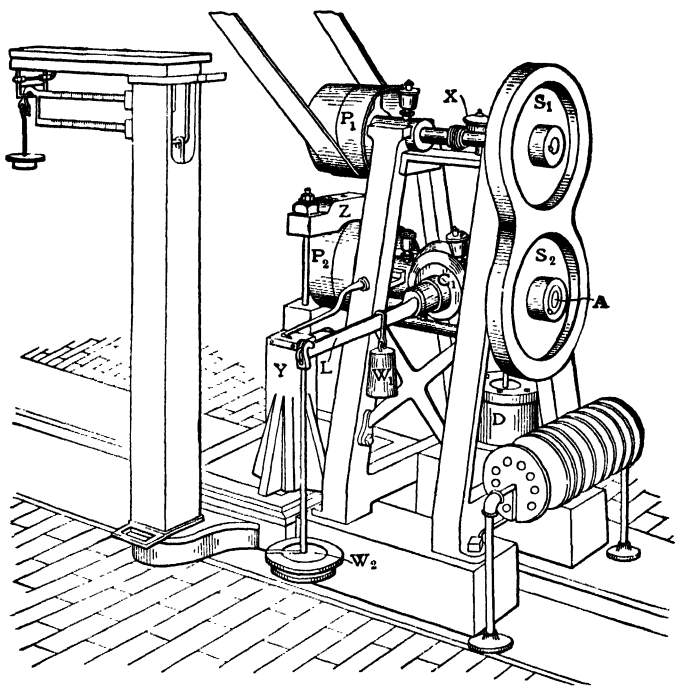


FIG. 175.—Webber differential-transmission dynamometer.

ment is graduated into 100 divisions and a bell is provided which rings at every 100 revolutions. Since the horsepower transmitted at a speed of **1 revolution** per minute is $\frac{\pi lw}{33,000}$ [Eq. (31)], then the horsepower corresponding to one division on the scale per 100 revolutions per minute is also $\frac{\pi lw}{33,000}$ for a perfect calibration.

Let v = the vertical force acting at C and C_1 (Fig. 174).

p = the vertical pressure between the teeth at each point of contact.

d = distance from the center of the rotation to C and C_1 .

l = the distance from the same center to the weight w .

Then from the foregoing discussion it should be clear that $2v = 4p$ and $wl = 2vd = 4pd$. If r is the effective pitch radius of the driving gear wheel B , r_1 is the radius of the small bevel wheels, and the force producing the turning movement in the shaft A is represented by f , we have

$$fr = 2pr_1,$$

and

$$f = \frac{2pr_1}{r} = \frac{wlr_1}{2dr}. \quad (32)$$

If we know the number of revolutions, then the space passed through by the force f can be calculated, and the work in foot-pounds is the product of the force times the distance passed through. The units given above are, of course, respectively, in feet and pounds.

About 10 horsepower can be transmitted and measured by one of these instruments having wheels B and D about 10 inches in diameter, when operating at 800 revolutions per minute.¹ Wear on the gears and noise become excessive at higher power and speed.

Calibration of a Differential Dynamometer.—1. Examine the dashpot and observe whether the piston moves freely in the cylinder, particularly without “sticking.” After the apparatus has been well oiled, the position of the poise to make the lever arm horizontal should be observed for “no load.” If this is not at zero, then all the readings on the scale must be corrected by the amount of this zero reading.

2. At each of the speeds required make a preliminary run without load and observe the reading of the poise when the lever is balanced.

3. Attach a Prony brake to the shaft from which the power is to be transmitted and observe for a series of loads and speeds the readings of the poise on the dynamometer lever.

4. For each speed plot a curve with theoretical **foot-pounds per minute** by Eq. (30) as abscissas and actual foot-pounds per minute as determined by the Prony brake as ordinates.

Torsion or Shaft Dynamometers.—When a shaft is subjected to a twisting moment, an angular twist is produced which is proportional to that moment. Thus if θ is the angle of twist produced by a twisting moment T in inch-pounds and if n is revolutions per minute, then

$$\text{b.hp.} = \frac{2\pi Tn}{12 \times 33,000} = cn\theta$$

where c is a constant determined by calibration with an absorption dynamometer.

Torsion meters, shown in Figs. 176, 177, 178, and 179, although applicable to large as well as small powers, have their most important

¹ WEBBER, S. S., and W. O. WEBBER, *Trans. A.S.M.E.*, Vol. 4, p. 277.

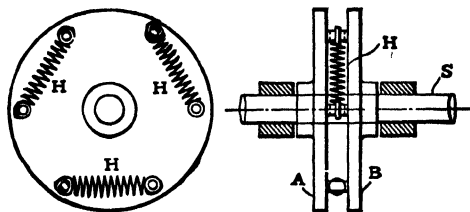


FIG. 176.—Spring-type shaft dynamometer.

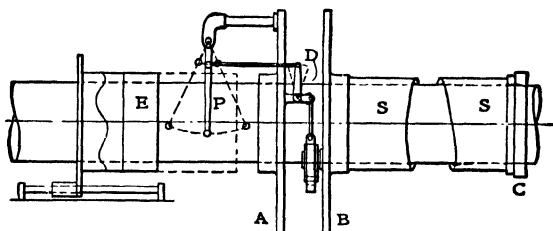


FIG. 177.—Mechanically operated shaft dynamometer.

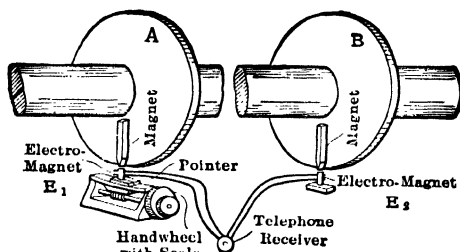


FIG. 178.—Electrically operated shaft dynamometer.

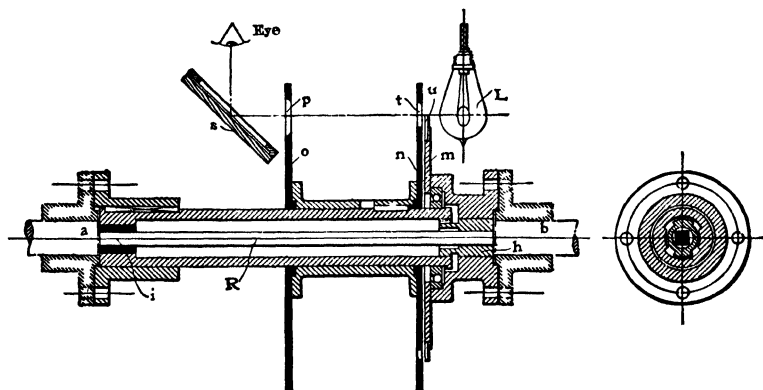


FIG. 179.—Shaft dynamometer with optical means of observation.

applications for measuring shaft horsepower of marine turbines and engines.

A **shaft dynamometer** consists essentially of a long metal tube encircling the shaft and fastened to it at one end, but free at the other, and is maintained in alignment by adjustable rollers; two radial arms, one attached to the shaft and the other to the free end of the tube, which rotate a slight amount with reference to each other, according to the twist of the enclosed length of shaft, and a set of levers which multiply this rotative movement and at the same time convert it into linear motion, which is transmitted to a sleeve and collar mounted upon the shaft and sliding thereon. These parts all revolve with the shaft. An independent indicating apparatus is provided, which is mounted on a stationary frame, and the sliding movements of the rotating collar are transmitted through it to an index hand. The torsional strain is determined from the reading of the accompanying scale, which is usually graduated in millimeters.

The zero reading is found by disconnecting the propeller and turning the shaft at a slow speed, first in one direction and then in the other, observing the indication in both cases, and fixing the point of zero strain at the mean of the two. When it is impracticable to disconnect the propeller, the readings may be taken when the vessel is drifting under her own headway after shutting off steam. The calibration of the instrument, which can best be done when the shaft is in the shop before installation, is carried on by securing the shaft in a fixed position and applying a torsional strain by means of weights at the end of a lever attached beyond the dynamometer, taking readings with a number of different weights.

The horsepower shown by the dynamometer is determined by multiplying the reading of the instrument expressed in millimeters by the number of revolutions of the shaft per minute and by a *constant determined from the calibration*. The constant is an expression for the horsepower corresponding to a speed of 1 revolution per minute and a reading of 1 millimeter.

A shaft dynamometer requires very delicate adjustment, as such instruments used on large steam turbines and engines requiring a shaft of comparatively large size have a movement at the end of the two arms of only 0.01 inch to produce a change of 500 horsepower in the load being transmitted.

The **Kenerson torsion dynamometer**¹ (Figs. 180 and 181) consists essentially of a divided shaft with a flanged coupling rigidly fastened to each of the adjoining ends. These flanges are only loosely connected by stud bolts and "latches." The latter are twisted by the power impressed so that their ends are forced against a pressure plate. The pressure against this plate is a measure of the power transmitted. Ball-

¹ *Trans. A.S.M.E.*, Vol. 31, pp. 171-179, 1909.

bearing races attached to a diaphragm covering an oil chamber communicate this pressure or thrust to a chamber in which the pressure is indicated by a Bourdon gage. Readings must be corrected for static head if the gage is placed above or below the couplings.

The **accelerometer** shown in Fig. 182 is used a great deal in automobile testing. From the indications of this instrument in terms of acceleration the power developed can be computed.¹ This instrument is designed on the principle that a pendulum hung on a moving body will be in a vertical position when at rest and moving uniformly. In the figure *D* is a copper disk pivoted on a rotating vertical axis, *M* is a magnet for damping the movement of this disk, and *G* shows two gear wheels of equal diameter,

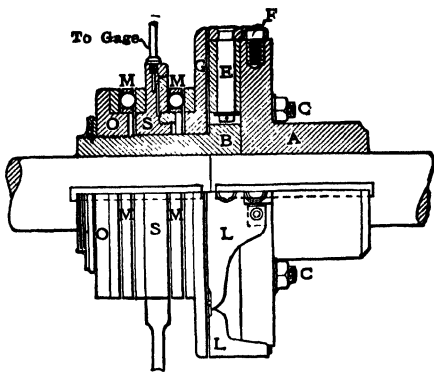


FIG. 180.—Kenerson dynamometer.

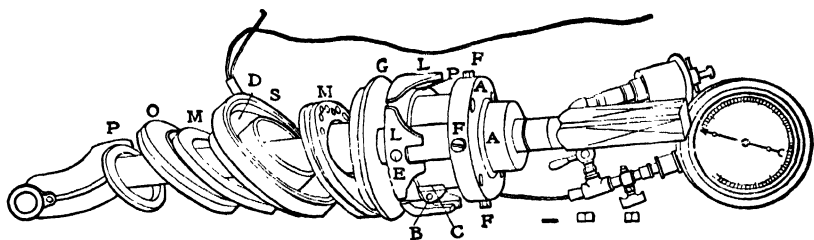


FIG. 181.—Parts of Kenerson dynamometer.

one fastened to the axis supporting the disk and the other on a separate axis carrying the needle *N*. A coil spring brings the needle back to zero.

One side of the disk *D* is heavier than the other, so that when the acceleration takes place in the direction of the arrow the heavier side tends to lag behind, causing a movement of the gears and the needle.

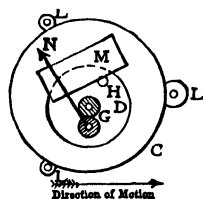


FIG. 182.—Accelerometer.

If *F* is the total resistance in pounds per ton at *N* miles per hour and *W* is the weight of the vehicle in tons, then brake horsepower equals $F \times W \times N$ divided by a constant.

Power Limits and Accuracy of Dynamometers.—The limits of speed and horsepower given in Table XIII A refer to the sizes made commer-

¹ *Internal Combustion Engineering* (London), Oct. 2, 1912, and *Engineering*, Sept. 16, 1910.

cially. Any of these types (with the exception of the Westinghouse turbine type) are made in sizes to absorb from 1 horsepower up. The probable error stated is approximate and refers to the apparatus in fair adjustment.

TABLE XIII A.—POWER LIMITS AND ACCURACY OF DYNAMOMETERS

Type	Approximate limit of speed, revolutions per minute	Usual power limit, horsepower	Probable error, per cent
Prony brakes:			
Block.....	1,000	10	1
Band.....	1,000	5	1
Wooden cleats on bands.....	1,000	100	1
Rope ($\frac{3}{4}$ -inch) with wooden cleats.....	1,000	50	1
Fluid friction dynamometers:			
Froude (ordinary "water" brake).....	2,500	500	1
Westinghouse (turbine type).....	4,000	5,000	0.2
Alden.....	300	3,000	1
Webb.....	3,000	200	1
Fan brake.....	2,000	200	1-5
Electric eddy-current brake.....	2,000	100	1
Electric generator.....	750-4,000	30,000	0.2
Transmission dynamometers:			
Webber.....	1,000	25	2
Torsion.....	1,000-3,000	50,000	2-5
Kenerson.....	1,500	100	2

Calibration of Platform Scales.—In Fig. 183 is shown the arrangement

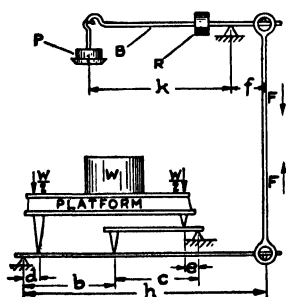


FIG. 183.—Diagram of platform scales.

of the levers of a platform scales. A large weight W is placed on the platform to be balanced by a small weight on the beam B and the poise R . If the weight P alone (with the poise R at zero) balances the weight W , then it follows, according to the law of moments, for the symbols given for dimensions in the figure, that

$$P \times \frac{k}{f} = F = \frac{W}{2} \times \frac{d}{h} + \frac{W}{2} \times \frac{e}{c} \times \frac{b}{h} \quad (33)$$

If the scales are constructed so that $\frac{e}{c} \times \frac{b}{h} = \frac{d}{h}$

then

$$P \times \frac{k}{f} = W \times \frac{d}{h}$$

$$\text{and the leverage ratio} = F = \frac{P}{W} = \frac{df}{kh}$$

In order to obtain the leverage ratio by direct measurement, the platform must be removed. Then measure carefully the distances between the knife edges of the various levers and also the length occupied by the 50 or 100 divisions on the graduated beam. One will save considerable time if these measurements are made with a decimal scale. From these data and Eq. (33), the leverage ratio can be obtained.

A better method of obtaining the leverage ratio consists of balancing a standard weight on the poise with a standard weight on the platform. Then from these data calculate the leverage ratio. An examination of the weights placed on the poise will show that these weights are marked with their actual weight and the weight they are supposed to balance on the platform. In case the scales do not balance in making this test, a little shot can be added to the poise or weight until a balance is obtained.

In calibrating platform scales against known weights, the scales should first be thrown out of adjustment; that is, the "adjusting" counterpoise (on the threaded rod of the graduated beam) should be moved from its present setting. To readjust the platform scales, the "adjusting" counterpoise is moved until a balance is obtained. This should be done without any weight on the scales. Standard weights now should be placed on the scale in 5-pound increments and the scales calibrated. The weights should be removed in a similar manner and the scales calibrated as the weights are removed. It is also well to calibrate the scales with weights placed at the four corners and compare these values with those obtained with weights at the center.

The *sensitiveness* of the scale is determined by finding the smallest added weight that will affect the balanced position of the beam first with no load and then with a 50-pound load upon the scales. This result will not be the same for different observers, as the results depend on how closely the beam is observed.

TABLE XIII B.—CAPACITY AND SENSITIVENESS OF WEIGHING DEVICES

	Usual capacity, pounds	Probable sensitiveness, pounds
Platform scales.....	100-2,000	$\frac{1}{16}$ - $\frac{1}{2}$
Spring balances.....	5-200	$\frac{1}{16}$ -1
Torsion balances.....	10	$\frac{1}{32}$ - $\frac{1}{4}$
Automatic scales:		
1. Pendulum type.....	50	$\frac{1}{32}$
2. Spring type.....	200	$\frac{1}{16}$
Chemical balances.....	50 (grams)	1 in 100,000

CHAPTER VII

FLOW OF FLUIDS

The flow of fluids will be discussed under these heads:

1. The flow of air.
2. The flow of steam.
3. The flow of water.

The Flow of Air.—When subjected to only a low pressure, air and many other gases are usually measured by a **gas meter**, of which there are many types sold commercially. There are, however, two general types:

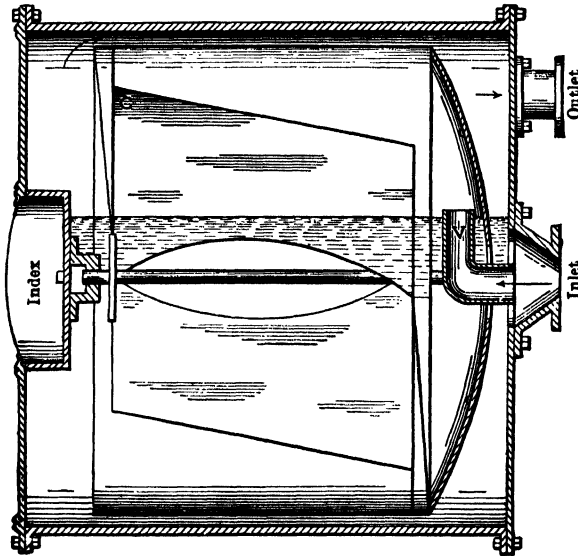


FIG. 184.—Typical "wet" gas meter.

(1) **wet** and (2) **dry**. The former is by far the more reliable and should be always used in preference to a "dry" meter when it can be obtained. "Wet" meters receive their name from the water seal maintained in them. This seal must be always kept at a constant level, determined by calibration, and before using such a meter in a test one should always observe whether the water level is at the standard mark. If it is not, then water must be added or withdrawn as the case may be. A section of a "wet" meter is shown in Fig. 184. It consists of a rotor somewhat resembling a paddle wheel revolving on a horizontal axis in an enclosed

cylindrical casing partly filled with water. Figure 185 illustrates a typical apparatus of this kind, with four compartments *A*, *B*, *C*, and *D*. When air or any gas flows¹ into one of the chambers of the meter, it accumulates over the surface of the water and by its pressure raises the chamber until it is filled.

During rotation, the central ports *a*, *b*, *c*, and *d* are opened and closed by means of the water seal. When these are open, gas from the pipe at the gas inlet is admitted to the compartments out of the water. Discharge ports *a'*, *b'*, *c'*, and *d'* through which the gas passes to the discharge pipe are also opened and closed by the water seal. If the drum revolves freely, no gas can pass through the meter without producing the required movement of the recording mechanism, because the admission and the

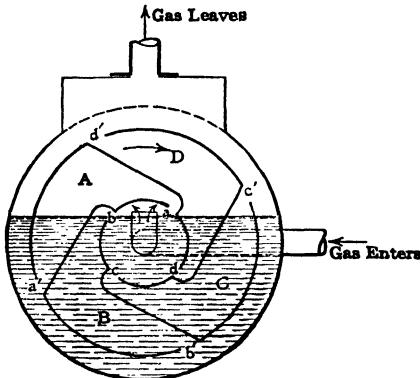


FIG. 185.—Diagram of "wet" gas meter.

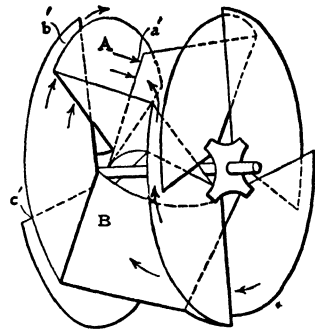


FIG. 186.—Rotor of "wet" gas meter.

discharge ports will not be open simultaneously in any compartment. Even the smallest rates of flow are accurately measured. Rotation is due to the difference in water level, as shown in the figure, between the two sides when the meter is operating. In the figure, gas is being admitted on the left-hand side so that the pressure on that side will be slightly greater than on the right, causing the water level to be higher and making a greater weight on the right-hand side than on the left. On account of the difficulty in making the admission and discharge ports of the rotor of sufficient size, the admission ports are usually placed at one end and the discharge ports at the other, as illustrated in Fig. 186, the flow of gas being shown by the arrows. In the assembled view (Fig. 184) the gas enters at the dry well *V*, passes through the drum and out at the front end, then over the drum between it and the inside of the meter casing to discharge at the outlet.² In this way the drum is made to revolve to the left by

¹ The nature or specific gravity of the gas is not important, as gas meters are calibrated to record volumes, usually cubic feet.

² More frequently the outlet for the gas is on the top of the casing than at the back as shown in the figure.

the pressure on the surface of the water below and the slanted partition *C* above, forming an ever-increasing pyramidal space between the surface of the water and the plane of the slanted partition.

Fluctuations of pressure or of velocity cause errors only when great enough to produce a sufficient surging of the water, so that the water sealing on the valves may at times be prevented. If the flow is intermittent, as in the suction pipe of gas engines and compressors, a pressure regulator must be provided. A rubber bag is used as a regulator when the gas is under pressure and a diving bell hung on springs for suction gas. A weighted gas regulator is shown in Fig. 233, page 245.

Dry gas meters are used for the usual "house metering" of gas. They are not nearly so accurate as the "wet" types but can be used more conveniently because they are not dependent on a constantly maintained water level. In simplest terms such meters consist of two chambers separated by a vertical partition, each chamber containing an

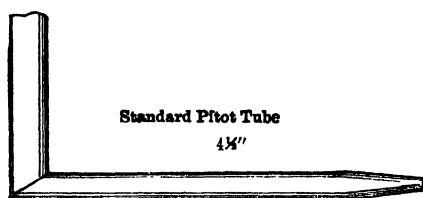


FIG. 187.—Standard Pitot tube (*ABC* type).

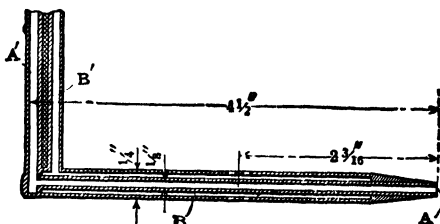


FIG. 188.—Section of Pitot tube.

interior measuring receiver having a flexible shell. Gas is admitted to these measuring receivers alternately by means of slide valves actuated automatically. The reciprocating movement of alternately filling and emptying these receivers operates the counting mechanism.

"Wet" meters are usually very accurate, while "dry" meters are not "supposed to be instruments of great accuracy."

Pitot Tube for Measurement of Air.—Probably the most accurate method of measuring air in large volumes is by means of a Pitot tube. A standard instrument of this kind designed for the measurement of air by the American Blower Corporation is shown in Fig. 187.

It consists simply of two concentric brass tubes, a small one *A* being placed inside of a larger one *B*, as illustrated in detail in Fig. 188. The lower end of the small tube is open at *A*, while the outside and larger tube is tapered and closed; but approximately midway along its horizontal portion, as shown in the figure, there are two holes on each side.¹ These holes should be not much more than $\frac{1}{16}$ inch in diameter. The

¹ The holes for static pressure are shown here at the top and bottom. In practice these holes are usually placed at the sides of the tube, as in this location they are less likely to become filled with dust and refuse.

tubes *A* and *B* are arranged so that each has a separate connection, as at *A'* and *B'*, as shown in Fig. 188.

The Taylor Pitot tube was formerly much used. It differs essentially from the one shown in the figures by having a short slot $2\frac{1}{2}$ inches long and $\frac{1}{16}$ inch wide on each side. At high velocities these slots cause eddies¹ and the static pressure observed will be too high. Taylor tubes, being 13 inches long compared with $4\frac{1}{2}$ inches for the *ABC* types, are much more clumsy and liable to breakage in handling.

As a Pitot tube is used, it is placed so that the opening at *A* (Fig. 188) points against the direction of flow and receives the full effect of the pressure due to the velocity of flow. The side openings in *B* are subjected to the static pressure only. For convenience, let p = velocity pressure and s = static pressure. For example, the difference in the levels in the manometer *a* (Fig. 190) is therefore that due to $(p + s) - s$, or simply p , the **velocity pressure**.

Another type of Pitot tube much used and known as **Burnham's** is shown in Fig. 189. It is made up of two tubes *A* and *B*. Tube *A* is for

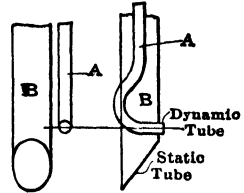


FIG. 189.—Burnham's Pitot tube.

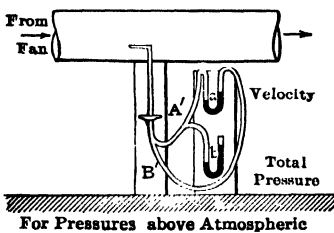


FIG. 190.

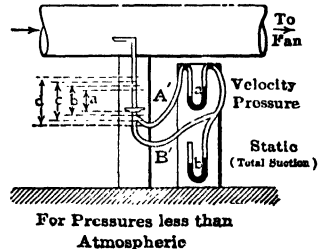


FIG. 191.

Arrangement of connections for Pitot-tube measurements.

obtaining the total pressure and in principle is not different from those already described. The static tube *B* is unique. It is open at the end and is pointed downward. On the side toward the direction of flow it is beveled at an angle of about 45 degrees. Neither the Taylor nor the Burnham type is satisfactory for measuring velocities above 6,000 to 8,000 feet per minute, while the *ABC* type is accurate for very high velocities. The latter type is the one recommended by the Power Test Committee of the A.S.M.E. (see *Mechanical Engineering*, November, 1912, page 1831).

¹ This inaccuracy of the Taylor type of tube can be remedied by soldering neatly a sheet of fine brass gauze over the slots. For results of tests with the Taylor tube see *Trans. A.S.M.E.*, Vol. 33, pp. 1137-1173.

Pitot tubes are usually connected to manometers or preferably to sensitive draft gages, showing the pressure in small fractions of an inch of water. When the end of the Pitot tube at A' is connected to the left-hand end of a draft gage, like those in Figs. 190 and 191, and the end at B' is attached to the right-hand end, the instrument acts as a differential gage and the difference between the reading when thus connected and its zero reading is the pressure in inches of water corresponding to the velocity alone; that is $(p + s) - s$. If, as before, we call the velocity pressure p in inches of water, and if h is the height or "head" in feet of an equivalent column of air producing the same pressure, then the velocity of the air v in feet per minute is

$$v = 60\sqrt{2gh},$$

where g is the force of gravity (32.2), and

$$\begin{aligned} h &= \frac{p}{12} \times \frac{\text{weight of a cubic foot of water}^1}{\text{weight of a cubic foot of air}} \\ &= \frac{62.3p}{12 \times \text{weight of a cubic foot of air}} = \frac{5.196p}{\text{weight of a cubic foot of air}}, \\ v &= 1,097\sqrt{\frac{p}{\text{weight of a cubic foot of air}}}. \end{aligned} \quad (34)$$

In Table XIV the weight is given of dry air and also the weight of air completely saturated with moisture (100 per cent humidity). The data given are at standard atmospheric pressure (29.92 inches of mercury) and the temperature given is that indicated by the "dry-bulb" thermometer (page 187). By interpolating between these tables, the weight of air for any temperature and degree of saturation is easily obtained. If it is remembered also that the weight per cubic foot is directly proportional to the absolute pressure, the weight for any pressure is readily determined. Tables for determining the percentage of saturation by means of wet- and dry-bulb thermometers are given on page 442.

For work requiring greater accuracy, the curve sheet of Fig. 287 (page 442) may be used, if it is remembered that for weight calculations 7,000 grains are equivalent to 1 pound.

To determine the volume of air flowing through a circular duct, the average velocity is most accurately obtained by dividing the cross section of the duct or pipe into 5 or 10 imaginary annular rings of **equal area**. Each of these rings is then again divided into two others of equal area. Observations for a given flow are then made by shifting the Pitot tube rapidly along a diameter and taking readings with the tip of the tube at each of these last points of subdivision, called "stations." Average velocities and flows are then readily obtained from these observations. This is sometimes known as the "10-point" method. For a

¹ The weight of a cubic foot of water at about "room" temperature (about 70° F.) is about 62.3 pounds.

pipe of radius r , divided into five annular rings, the "stations" or points of observation would be at the following distances from the center: (1) $0.316 r$; (2) $0.548 r$; (3) $0.707 r$; (4) $0.837 r$; (5) $0.949 r$.

TABLE XIV.—PROPERTIES OF SATURATED AIR¹

Weights of Air, Vapor of Water, and Saturated Mixture of Air and Vapor at Different Temperatures, under Standard Atmospheric Pressure of 29.921 Inches of Mercury

Temperature, degrees Fahrenheit	Weight in a cubic foot of mixture			B.t.u. absorbed by 1 cu. ft. saturated air per degree Fahrenheit	Cubic feet saturated air warmed 1° per B.t.u.	Specific heat, B.t.u. per pound of mixture
	Weight of dry air, pounds	Weight of vapor for saturation, pounds	Total weight of saturated mixture, pounds			
(1)	(2)	(3)	(4)	(5)	(6)	(7)
0	0.08625	0.000068	0.08632	0.02083	48.02	0.2413
5	0.08528	0.000087	0.08537	0.02060	48.54	0.2414
10	0.08433	0.000110	0.08444	0.02039	49.05	0.2415
15	0.08339	0.000139	0.08353	0.02018	49.56	0.2416
20	0.08246	0.000176	0.08264	0.01998	50.07	0.2418
25	0.08154	0.000222	0.08176	0.01978	50.57	0.2419
30	0.08062	0.000277	0.08090	0.01958	51.07	0.2420
35	0.07970	0.000339	0.08004	0.01939	51.58	0.2423
40	0.07878	0.000409	0.07919	0.01921	52.06	0.2426
45	0.07786	0.000491	0.07835	0.01903	52.55	0.2429
50	0.07694	0.000587	0.07753	0.01885	53.05	0.2431
55	0.07601	0.000699	0.07671	0.01868	53.54	0.2435
60	0.07506	0.000828	0.07589	0.01851	54.02	0.2439
65	0.07409	0.000978	0.07507	0.01835	54.50	0.2444
70	0.07310	0.001151	0.07425	0.01819	54.97	0.2450
75	0.07208	0.001350	0.07343	0.01804	55.44	0.2457
80	0.07103	0.001578	0.07261	0.01790	55.87	0.2465
85	0.06993	0.001838	0.07177	0.01775	56.34	0.2473
90	0.06879	0.002134	0.07092	0.01762	56.76	0.2485
95	0.06760	0.002470	0.07007	0.01749	57.18	0.2496
100	0.06635	0.002850	0.06920	0.01736	57.59	0.2509
105	0.06503	0.003279	0.06831	0.01725	57.98	0.2525
110	0.06364	0.003762	0.06740	0.01714	58.35	0.2543
115	0.06217	0.004305	0.06648	0.01704	58.69	0.2563
120	0.06060	0.004914	0.06551	0.01695	59.00	0.2587
125	0.05893	0.005594	0.06452	0.01686	59.31	0.2613
130	0.05715	0.006351	0.06350	0.01679	59.56	0.2644
135	0.05524	0.007191	0.06243	0.01673	59.78	0.2680
140	0.05319	0.008120	0.06131	0.01668	59.96	0.2721
145	0.05099	0.009153	0.06014	0.01664	60.11	0.2767
150	0.04864	0.010295	0.05894	0.01662	60.17	0.2820
155	0.04612	0.011553	0.05767	0.01661	60.21	0.2880
160	0.04340	0.012936	0.05634	0.01662	60.17	0.2950
165	0.04048	0.014451	0.05493	0.01663	60.14	0.3028
170	0.03734	0.016108	0.05345	0.01668	59.96	0.3121
175	0.03397	0.017919	0.05189	0.01675	59.71	0.3228
180	0.03035	0.019896	0.05025	0.01684	59.38	0.3351
185	0.02646	0.022053	0.04851	0.01695	59.00	0.3495
190	0.02228	0.024400	0.04668	0.01710	58.49	0.3663
195	0.01780	0.026950	0.04475	0.01730	57.81	0.3866
200	0.01300	0.029715	0.04272	0.01749	57.18	0.4094
205	0.00783	0.032707	0.04054	0.01767	56.50	0.4359
210	0.00230	0.035938	0.03824	0.01802	55.50	0.4712
212	0.0	0.037307	0.03731	0.01815	55.10	0.4865

¹ Buffalo Forge Company, "Fan Engineering," 3d ed., p. 15, 1933.

When the Burnham Pitot tube is used, readings are taken usually in only one position. This position giving the average velocity is stated to be at a distance of eight-tenths of the actual internal radius from the center of the pipe.

The following paragraph is the recommendation of the American Society of Heating and Ventilating Engineers:

In general, no accurate velocity-pressure readings can be taken when the flow of air in ducts is turbulent. To insure accuracy, a straight section of duct from five to ten times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work, it is necessary to make a traverse of the duct, dividing its cross section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

Figures 190 and 191 show the methods of connecting a Pitot tube to

manometers for observing velocities when the pressure is above or below atmospheric. The usual case is where the pressure is greater than atmospheric, and the cases where it is less are most often in the suction line of a ventilating fan. In Fig. 191 positions of "stations" (*a*, *b*, *c*, and *d*) are marked on a board to assist in taking observations. Measurements of velocity with a Pitot tube should not be attempted if there is not at least 15 feet of straight pipe in the direction in which the tube is pointed. This precaution is necessary to avoid the effect of eddies in the pipe.

Anemometers.—A very convenient and simple method of measuring directly the volume of air, or any gases, is by using an instrument called an anemometer.

This instrument (Fig. 192) consists in its essential parts of a light vane wheel like a screw propeller having either flat or slightly curved vanes mounted on slender arms. The wheel must be made very light in weight, must be accurately balanced, and should move easily in its bearings. By its own motion it operates a counting mechanism attached to its shaft which indicates velocities in feet. Read-

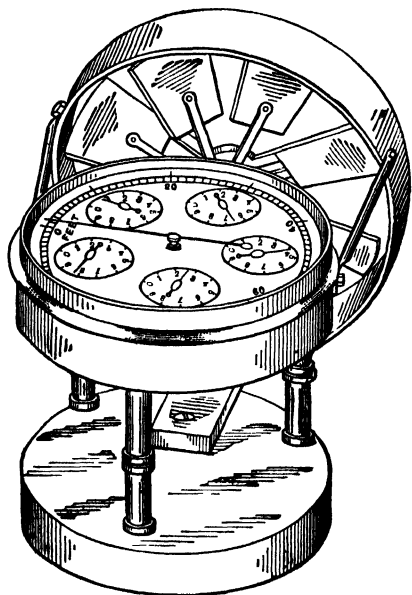


FIG. 192.—A typical anemometer for measuring velocity of air.

ings of the counter are taken at the beginning and end of a suitable lapse of time, usually $\frac{1}{2}$ to 1 minute. Such instruments must be placed with the axis of rotation in the direction of the flow of air or gas. They have upper and lower velocity limits beyond which they should not be used. The lower limit cannot be defined, as it will depend on the precautions taken in manufacture and in use to eliminate friction. As regards the higher limit, it will usually depend on the size of the wheel, large wheels being less suitable than smaller ones for high velocities. Practically none should be used for velocities higher than 1,000 feet per minute.

A calibration chart must always be provided for such instruments and the calibration should be frequently checked at several points within its velocity limits.

Methods of Calibrating Anemometers, Pitot Tubes, and Gas Meters.—

Probably the best method of calibrating anemometers is to compare their readings with the actual measurements of air discharged from a gasometer or gas holder like Fig. 193. It consists of a tank *A* for holding water or other liquid into which the "bell" *B* is raised and lowered. The piping as shown is arranged with a three-way cock (see page 149), but for accurate work separate inlet and discharge pipes should be provided. The weight *W* is to counterbalance the weight of the bell. This method of counterbalancing, if used without a means for correction, would cause a change of pressure of the gas in the holder as the bell ascends or descends, due to a variation of depth of immersion. As a means to correct this, a compensating weight *w* is suspended from a cord wrapping over a can *C*. Observations required to determine the volume of gas are: (1) pressure (usually read with a water manometer *M*); (2) temperature; and (3) movement of the bell in a given time. In order to get the temperature of the gas in the bell accurately, the temperature of the liquid should be as nearly as possible the same as that of the gas. Movement of the bell is usually read directly with the help of a suitable scale and some form of sighting device to insure accuracy in reading the scale.

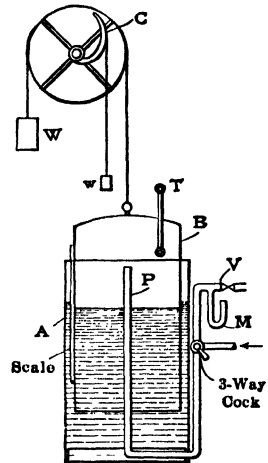


FIG. 193.—Gasometer.

Gasometers may be calibrated by calculation or by actual tests, preferably of the displacement of water. When the method of calculation is used, measurements of the diameter should be made at several points on the circumference to allow for possible lack of symmetry.

Large gasometers such as are installed at gas works are also frequently used to calibrate Pitot tubes; and gas meters are invariably calibrated by comparison with a gasometer.

Gas meters may also be calibrated by any apparatus suitable for the displacement of the gas as it is withdrawn by water or other suitable liquid. It is necessary, of course, that when the weighings are made the pressure and temperature of the gas be accurately determined.

Anemometers are suitable only for low velocities (from about 50 to 1,000 feet per minute) and Pitot tubes in general are best adapted to velocities from 300 to 3,000 feet per minute, but instruments like the *ABC* type (Fig. 187), designed to avoid eddies around the "static" openings, are satisfactory up to 6,000 feet per minute.

In order to use an anemometer successfully, all the gas to be measured must be passed through openings of suitable size in which the instrument can be placed. These openings should each have an area of about 15 square inches (if the anemometer is about $2\frac{1}{2}$ inches in diameter) so that the resistance interposed by the instrument will be negligible.

A very common method of calibrating anemometers and Pitot tubes is by mounting them on the end of a long and light rod arranged to be revolved about a central point. The readings of the instrument are compared with its computed velocity. Other methods of calibration under more nearly "working conditions" are generally considered better, as the method of swinging about a central point makes the instrument read too high on account of the eddies produced.

The standard of reference for calibrating Pitot tubes, anemometers, etc., is the gasometer,¹ and if the instrument used cannot be calibrated by actual measurement, the constants employed should be those obtained from a similar instrument which has been calibrated by actual reference to a gasometer measurement. (Report of Power Test Committee, A.S.M.E., 1912.)

Flow of Air through an Orifice.—Air under comparatively high pressures is usually measured in practice by means of pressure and temperature observations made on the two sides of a sharp-edged orifice in a diaphragm. Figure 194 illustrates the method with two pressure gages on opposite sides of the orifice and a thermometer for obtaining the temperature t_1 at the initial or higher pressure p_1 . The flow of air w , in pounds per second, may then be calculated² by **Fliegner's formulas**.

¹ It is very difficult to get a uniform temperature in a gasometer so that the volume of gas discharging from the outlet may be different by 2 or 3 per cent from that indicated by the scale on the gasometer. When accurate work is to be done, a better method to use is the displacement of water as measured by its weight in a vessel of heavy sheet metal. The walls of the vessel should be heavy to prevent rapid radiation to the surrounding air. A long thermometer should be used which should be inside the vessel itself and may be read through a peep hole. If the vessel is of heavy glass, the conditions will be still better.

² Coefficients of discharge must be applied when the formulas are used in connection with actual data in the same way as in Eq. (37) below. Research at the University of Missouri by Professor Westcott gives the value of 0.925 for this coefficient.

$$w = 0.530 \times a \frac{p_1}{\sqrt{T_1}} \text{ when } p_1 \text{ is greater than } 2p_2. \quad (35)$$

$$w = 1.060 \times a \sqrt{\frac{p_2(p_1 - p_2)}{T_1}} \text{ when } p_1 \text{ is less than } 2p_2, \quad (36)$$

where a is the area of the orifice in **square inches**, T_1 is the absolute initial temperature in degrees Fahrenheit at the absolute pressure p_1 in the "reservoir or high-pressure side," and p_2 is the absolute discharge pressure, both in pounds per **square inch**. When the discharge from the orifice is directly into the atmosphere, p_2 is obviously barometric pressure.

For small pressures it is often desirable to substitute manometers for pressure gages. One leg of a U-tube manometer can be connected to the high-pressure side of the orifice and the other leg to the low-pressure side. Many engineers insert valves or cocks between the manometer and the pipe in which the pressure is to be observed for the purpose of "damping" oscillations. This practice is not to be recommended, as

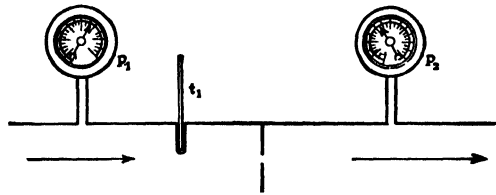


FIG. 194. —Measuring flow of air through an orifice.

there is always the possibility that the pressure is being throttled.¹ A better method is to use a U-tube made with a restricted area at the bend between the two legs. This will reduce oscillations and not affect the accuracy of the observations.

Discharge from compressors and the air supply for gas engines are frequently measured by orifice methods.

When $p_1 - p_2$ is small compared with p_1 , the simple law of discharge² of fluids can be used as follows:

¹ Report of Power Test Committee, *Mech. Eng.*, November, 1912, p. 1695.

² If the density is fairly constant,

$$\frac{144p_1}{s} + \frac{v_1^2}{2g} = \frac{144p_2}{s} + \frac{v_0^2}{2g},$$

where v_1 is the velocity in feet per second in the "approach" to the orifice and v_0 is the velocity in the orifice itself. Since v_1 should be very small compared with v_0 ,

$$\frac{v_0^2}{2g} = \frac{144(p_1 - p_2)}{s},$$

$$v_0 = \sqrt{\frac{2g \times 144(p_1 - p_2)}{s}}.$$

$$w = \frac{f a v_0 s}{144} = f a s \sqrt{\frac{2g \times 144(p_1 - p_2)}{s}},$$

$$w = \frac{f a}{144} \sqrt{2g \times 144(p_1 - p_2)s}.$$

$$w = \frac{fa}{144} \sqrt{2g \times 144(p_1 - p_2)s}, \quad (37)$$

where w is the weight in pounds discharged per second, a is the area of the smallest section of the orifice in square inches, the pressures p_1 and p_2 are in pounds per square inch, f is a coefficient from experiments, g is the acceleration due to gravity (32.2), and s is the unit weight of the gas measured, in pounds per cubic foot, for the average of the initial and final conditions of temperature and pressure (table of weight of air on page 187). If the difference in pressure is measured in inches of water h with a manometer, then in terms of pounds per square foot

$$144(p_1 - p_2) = \frac{62.4}{12} \times h$$

$$w = \frac{fa}{144} \sqrt{2ghs \times \frac{62.4}{12}} \text{ pounds per second,}$$

where 62.4 is the weight of a cubic foot of water (density) at usual "room" temperatures.

This equation can also be transformed so that a table of the weight of air is not needed, since by elementary thermodynamics $144pv = 53.3T$, where v is the volume in cubic feet of 1 pound of air or other gas and T is the absolute temperature in degrees Fahrenheit. Since v is the reciprocal of s , then

$$s = 144p \div 53.3T,$$

and

$$w = 0.209fa \sqrt{\frac{hp}{T_1}}. \quad (38)$$

Here p and T should be the values obtained by averaging the initial and final pressures and temperatures. Great care should be exercised in obtaining correct temperatures. When Eq. (37) is used, for accurate work, corrections of s for humidity must be made.¹

For measurements made with orifices with a well-rounded entrance and a smooth bore, so that there is practically no contraction of the jet, the coefficient f in Eqs. (37) and (38) may be taken as 0.98. In the rounding portion of the entrance to such a nozzle the largest diameter must be at least twice the diameter of the smallest section. For circular orifices with sharp corners, Professor Dalby² in reporting extensive experiments stated that the coefficient for his sharp-edged orifices in a thin plate of various sizes from 1 to 5 inches in diameter was in all cases approximately 0.60; and these data agree very well with those published by Durley.³

¹ Tables of relative humidity and moisture in air are given on p. 442.

² *Engineering* (London), Sept. 9, 1910, p. 380, and Ashcroft in *Proc. Inst. Civil Eng.*, Vol. 173, p. 289.

³ *Trans. A.S.M.E.*, Vol. 27, p. 193.

When $p_2 \div p_1 = 0.99$, the values obtained with this coefficient are in error less than $\frac{1}{2}$ per cent; and when $p_2 \div p_1 = 0.93$, the error is less than 2 per cent.

Flow of Air Measured by Cooling.—This method depends on taking from the air an amount of heat¹ which can be measured and then computing from the heat units absorbed, the difference in temperature and specific heat of the air, its weight and volume.² The arrangement of the apparatus is shown in Fig. 195. A coil of pipes *C*, the cooling surface of which is as equally as possible distributed over the section of the duct *D*, *D'*, carrying the air to be measured, is used to absorb heat

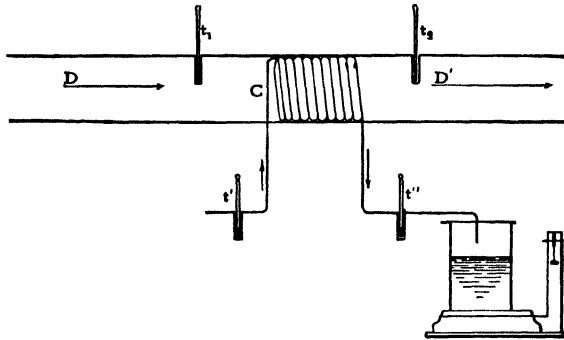


FIG. 195.—Measuring flow of air by cooling.

by circulating water through it. Thermometers are arranged so that the temperatures of the air and of the water can be observed, and a platform scale is shown for obtaining the weight of water. By using the symbols t_1 and t_2 for the initial and final temperatures of the air in degrees Fahrenheit, t' and t'' for the temperatures of the water entering and leaving in degrees Fahrenheit, w_a = weight of air passing through duct in pounds per second, w_0 = weight of water collected in pounds per second, and 0.2375 = specific heat of the air at constant pressure and at temperatures not much above "atmospheric," then the heat absorbed by the water per second is $w_0(t'' - t')$ B.t.u. and this equals the heat lost by the air, or $0.2375 w_a(t_1 - t_2)$, and therefore

$$w_a = \frac{w_0(t'' - t')}{0.2375(t_1 - t_2)} = \frac{4.211w_0(t'' - t')}{(t_1 - t_2)}. \quad (39)$$

Thomas's electric meters employ this method of measuring gas, using electric current for the heating medium. The gas flows past a series of

¹ The method will be equally applicable if heat is added, as, for example, by passing steam through the coil. This method is often used to calibrate Pitot tubes and anemometers.

² The weight of a cubic foot of air is 0.0765 pound at 62°F. (522°F. abs.) and 14.7 pounds per square inch pressure. Since the volume is directly proportional to the absolute temperature, the weight at any other temperature is easily computed.

electrical resistances enclosed in a pipe. Electric heating by these resistances raises the temperature of the gas a few degrees. Temperatures before and after heating are measured by delicate resistance thermometers connected to a Wheatstone bridge and a sensitive galvanometer.

If the electrical energy used for heating is kept constant, and if the specific heat of the gas does not vary, the flow is inversely proportional to the rise in temperature.

If E is the amount of energy in watts, supplied to raise the temperature of w pounds of gas per hour through $t^\circ\text{F.}$ and s is the specific heat of the gas at constant pressure, then

$$w = \frac{3.412E}{ts}. \quad (40)$$

If the volume of gas is required, its pressure, temperature, and relative humidity (percentage of saturation) must be determined.

Receiver Method of Measuring Air.—None of the preceding methods are adaptable for measuring the volume of air at high pressures as in the case of measuring the discharge in tests of air compressors. Pumping air into a suitably strong receiver is a method often used. The compressor is made to pump against any desired pressure which is kept constant by a regulating valve on the discharge pipe:

P_1 and P_2 = absolute initial and final pressures for the test, in pounds per square inch.

T_1 and T_2 = mean absolute initial and final temperatures, in degrees Fahrenheit.

W_1 and W_2 = initial and final weight of air in the receiver, in pounds.

V = volume of receiver, in cubic feet.

$P_1V = WRT_1$ and $P_2V = W_2RT_2$, where R is the constant 53.3 for air, then weight in pounds of air pumped,

$$W_2 - W_1 = \frac{V}{53.3} \left(\frac{P_2}{T_2} - \frac{P_1}{T_1} \right). \quad (41)$$

In accurate laboratory tests the humidity of the air entering the compressor should be measured in order to reduce this weight of air to the corresponding equivalent volume at atmospheric pressure and temperature.

Principal errors in this method are due to difficulty in measuring the average temperature in the receiver. Whenever practicable, the final pressure should be maintained in the receiver at the end of the test until the final temperature is fairly constant.

The foregoing method is often reversed by discharging air at high pressure from a receiver. Constant discharge pressure is maintained by throttling with a valve.

Measurement of Air by Chemical Analysis.—Quantity of air supplied for combustion in boilers and other furnaces can be determined from the analysis of the products of combustion by the formulas given on pages 317 and 330.

Venturi meters (see page 211) are also used successfully for measuring large volumes of gas as in tests of gas producers,¹ etc.

Flow of Steam through Nozzles and Orifices.—The flow of the steam from an orifice or nozzle has a definite critical value when the final absolute pressure is approximately 0.58 of the initial pressure. When the final pressure is less than this critical value, the flow is expressed very accurately by the following empirical formula, based on the experiments of Professors Emswiler and Fessenden. Using the following symbols:

p_1 = initial absolute pressure of the steam, in pounds per square inch.

p_2 = final absolute pressure of steam, in pounds per square inch.

a = area of the smallest section of the nozzle or orifice, in square inches.

Then the weight of the dry saturated steam discharged in pounds per second is approximately²

$$w = \frac{p_1^{.97}a}{60} \text{ when } p_2 \text{ is less than } 0.58p_1. \quad (42)$$

Now since in the theoretical formulas the weight discharged is inversely proportional to the square root of the specific volume v , or w is proportional to $\sqrt{\frac{p_1}{v}}$, the formula above corrected for initial quality x of the steam is

$$w = \frac{p_1^{.97}a}{60\sqrt{x}} \text{ when } p_2 \text{ is less than } 0.58p_1.^3 \quad (44)$$

When the steam is **superheated**, the specific volume is considerably increased, and for this condition the author has found that the following equation gives very satisfactory results:⁴

$$w = \frac{p_1^{.97}a}{60(1 + 0.00065d)}, \quad (45)$$

¹ *Trans. A.S.M.E.*, Vol. 28, p. 483, and Vol. 29, p. 952. See also *Bull.* 76, Builder's Iron Foundry, Providence, R. I.

² A somewhat simpler formula, known as **Napier's formula**, which is accurate enough for most calculations, is the following:

$$w = \frac{p_1a}{70} \text{ when } p_2 \text{ is less than } 0.58p_1. \quad (43)$$

³ Emswiler states the correction factor for quality of steam as $1 \div (1 - 0.012m)$ where m is the per cent of moisture in the steam.

⁴ For a more extended discussion of the flow of steam see "Steam Turbines," by the author, 6th ed., pp. 36-45.

when, as before, p_2 is less than $0.58p_1$ and where d is the number of degrees (Fahrenheit) of superheat.

When the final pressure p_2 is greater than $0.58p_1$, the formulas must be modified to correspond to the reduced flow observed by inserting a coefficient K as a factor in the right-hand member of the equations. Values of this coefficient are most conveniently obtained from the curve in Fig. 196, which was plotted from the experimental results obtained by Professor Rateau.

Equations (42) to (45) are for the flow through nozzles with smooth walls, being well rounded at the entrance and the length along the axis at least three times the length of the shortest side or diameter. If the nozzles are of approximately rectangular section, they must be made

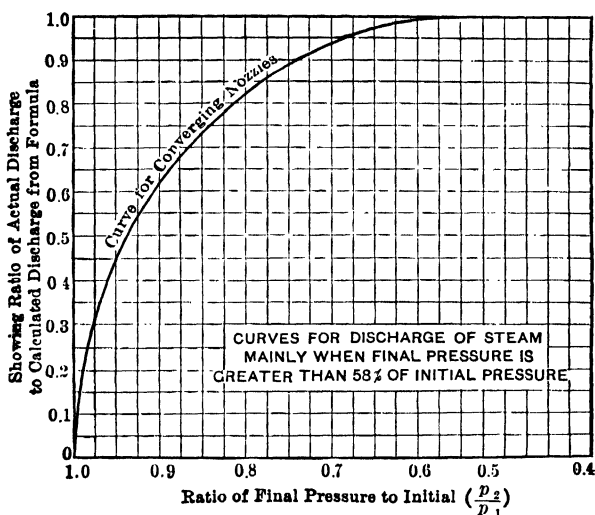


Fig. 196.—Rateau's curve for flow of steam giving values of the coefficient K .

without well-defined edges; in other words, the cross section must show well-rounded corners.

Orifice measurements of the flow of steam are particularly recommended by some engineers for ascertaining the steam consumption of the "auxiliaries" in a power plant. This method commends itself particularly because of its simplicity and accuracy. It is best applied by inserting a plate $\frac{1}{8}$ inch thick with an orifice 1 inch in diameter, with **square** edges, at its center, between the two halves of a pair of flanges on the pipe through which the steam passes. Accurately calibrated steam gages are required on each side of the orifice to determine the loss of pressure. The weight of steam for the various differences of pressure may be determined by arranging the apparatus so that the steam passing through the orifice will be discharged into a tank of water placed on a platform scales. The flow through this orifice in pounds of dry saturated

steam per hour when the discharge pressure at the orifice is 100 pounds by the gage is given by the following table:¹

Pressure, drop, pounds per square inch	Flow of dry steam per hour, pounds	Pressure, drop, pounds per square inch	Flow of dry steam per hour, pounds
$\frac{1}{2}$	430	5	1,560
1	615	10	2,180
2	930	15	2,640
3	1,200	20	3,050
4	1,400		

Measurement of Steam with Nozzles for Turbine-performance Tests (Requirements of A.S.M.E. Research Committee).²—A system of steam measurements by the use of nozzles has recently been worked out with sufficient accuracy to make it adaptable to determine the "guarantee" performance of steam turbines for noncondensing operation, for condensing operation with jet condensers, and of extraction or bleeder steam turbines. Engineers have realized for a long time that a reasonably accurate and standardized method was needed for determining accurately the steam consumption of a steam turbine by the method of measuring the pressures within and near to a suitably designed orifice in the steam supply line. The Steam Turbine Test Code (page 380) of the A.S.M.E., as now published, makes no provision for determining steam quantities by orifice-flow measurements. The American practice in this respect is different from the European, as the Steam Turbine Rules for Acceptance Tests of the International Electrotechnical Commission permit such determinations by means of standardized nozzles. Such nozzle measurements must not be confused, however, with the very common use of commercial types of steam-flow meters. In respect to the latter, turbine manufacturers have been justified in the past when they have declined to accept for guarantee determination results which were based on data obtained with commercial flow meters. It must be realized, however, that in some cases the arrangement of the plant is such that a test to determine steam consumption cannot be carried out by the method of weighing the condensate without excessive expense. There is, therefore, a need for a less costly measurement of steam than by weighing.

Typical Flow Nozzles.—Figures 197 to 200 show recommended types of flow nozzles designed by S. A. Moss of Lynn for accurate steam-flow measurements. Of these, the nozzle shown in Figs. 197 and 199 (with pipe taps) is especially recommended. In this nozzle, the approach side

¹ *Jour. A.S.M.E.*, November, 1912, p. 1693.

² Moss, S. A., and W. W. JOHNSON, *Trans. A.S.M.E.*, Vol. 55, No. 11, 1933.

is well rounded. As this nozzle is shown in cross section, each side is practically a quarter ellipse with a ratio of the axes of 3 to 2.¹

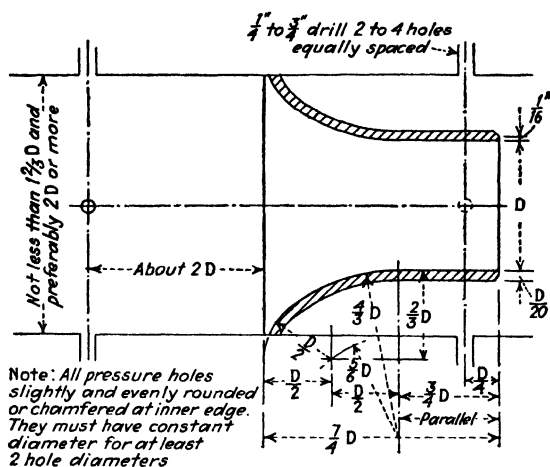


FIG. 197.—Recommended nozzle with pipe taps.

Even though, as in the nozzle described, there is a very gradual curve of approach, there can be no certainty that the curved portion of the nozzle will be completely filled by the flow of steam through it.

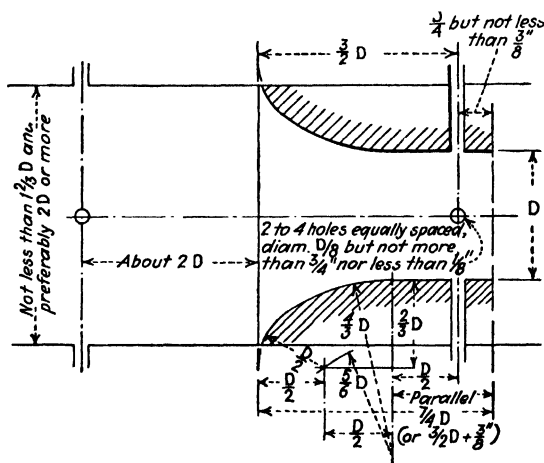


FIG. 198.—Recommended nozzle with throat taps.

For this reason, a short portion of a nozzle used for accurate flow measurements should have parallel sides, so as to form a nozzle throat, of which

¹ This cross section is nearly the shape used by the Bureau of Standards for determining discharge coefficients for the flow of air (*Research Paper 49*, March, 1929).

the length should be at least half of the smallest diameter of the nozzle. Another reason for having this portion of the nozzle with parallel sides is to insure that the jet of steam flowing through the nozzle has the exact diameter of the nozzle at the point where its static pressure is taken, as, for example, at the drill holes shown at the right-hand side of Figs. 197, 198, and 199. When this *uncertainty* as to the size of the jet of the flowing steam is thus removed, the *nozzle-discharge coefficient* can be very accurately determined.

In the application of flow nozzles for accurate steam measurements, the nozzle to be used must be selected to give, for all conditions of the test, differential pressures (page 19) of at least 2 inches of mercury¹ for the smallest flow to be measured, and between 10 inches of mercury and 10 per cent of the "high-side" pressure for the largest flow. In a good

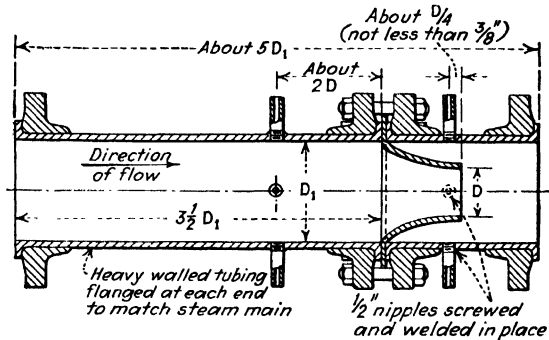


FIG. 199.—Special pipe section for nozzle with pipe taps.

design of flow nozzle, the throat diameter D should be between 40 and 60 per cent of the steam-pipe diameter. For determining the best nozzle diameter for a given case, formula (47) on page 203 may be used.

Static-pressure Holes in Nozzle.—The nozzles that have been described are intended to have either two or four low-side static-pressure holes. Nozzles that have a throat diameter of 2 inches or more should have throat-pressure holes (if any) that are not less than $\frac{1}{4}$ inch but not more than $\frac{3}{4}$ inch in diameter. For nozzles smaller than those with a 2-inch throat diameter, the throat-pressure holes should have a diameter of about $\frac{D}{8}$ inch but should not be smaller than $\frac{1}{8}$ inch. All static-pressure holes should have a straight length of at least twice the hole diameter. These static-pressure holes should be located at a distance of about $\frac{D}{4}$

¹ Pressure loss due to friction through the nozzle can, of course, be minimized by selecting a nozzle that will give a small differential pressure, but this is undesirable, as it is best to use the differential pressures as stated, in order to obtain satisfactory accuracy in the determination of the differential pressure.

inch but not less than $\frac{3}{8}$ inch upstream from the plane of the nozzle end. They may be arranged in either of two methods: (1) as throat-pressure holes, which, of course, would be directly in the nozzle throat, as shown in

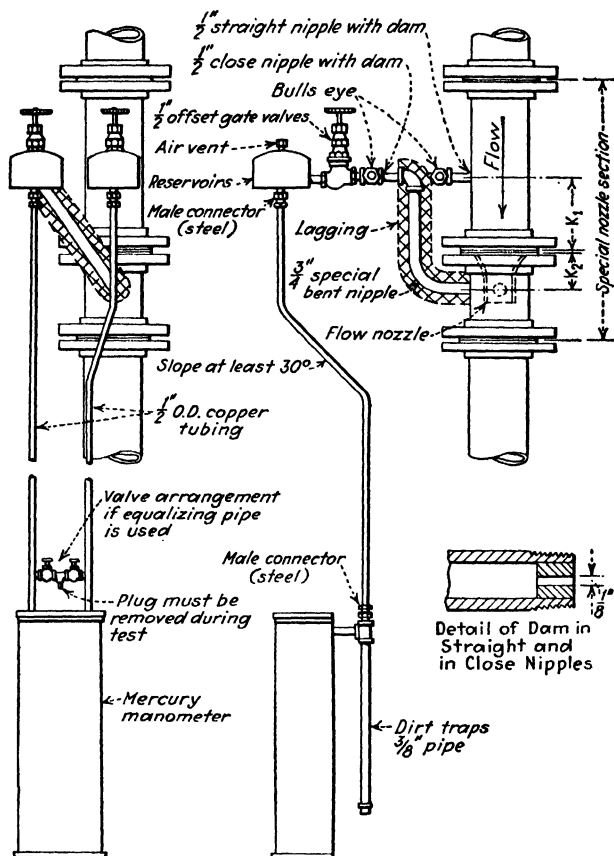


FIG. 200.—Layout of apparatus for measuring nozzle with pipe taps in vertical steam main.

Fig. 198, so as to give the actual throat pressure;¹ or (2) as holes in the pipe wall, as shown in Figs. 197 and 199, so as to give the pressure in the region into which the jet discharges.

¹ When throat taps are used, special care must be taken, on account of the high throat velocity, to insure that the inner edges are perfectly formed. This may be accomplished by first rough-finishing the nozzle throat, drilling, reaming, and then plugging the throat holes with tight plugs, and finally finishing the throat; or the nozzle throat may be finished by grinding after the holes are drilled. In either case, the inner edges of the pressure holes should be smoothed with crocus cloth so as to remove all sharp edges and burrs. These edges should be inspected with a magnifying glass.

For accurate work in nozzle flow, it is necessary for the nozzle to be accurately centered in the pipe in which the flow is measured. A nozzle should therefore be doweled or similarly secured in place so that there is no opportunity for the center line of the nozzle throat to deviate from the center line of the pipe. A small amount of eccentricity of the nozzle with respect to the center line of the pipe may produce sizable errors. This system of steam measurement, as thus far developed, applies only to cases where the steam preceding the nozzle is superheated. In order to determine the superheat, the temperature of the steam should be measured with two calibrated mercury-in-glass thermometers inserted in finned wells (page 39) or by first-class resistance thermometers. In either case, the glass or resistance thermometers should be located in the pipe ahead of the nozzle at distances from it of not less than $16D$ inches, and preferably $20D$ inches. Thermometer wells in a vertical pipe should be located at an angle of about 45 degrees, the finned thermometer wells shown on page 40 being used.

Calibration to Determine Nozzle-discharge Coefficient.—When a special separate nozzle section is used, the nozzle-discharge coefficient may be determined by calibration of the complete special section in a laboratory by the actual weight of the steam flow through the nozzle, the differential pressure at the nozzle being measured through the static holes in the same way that static pressures will be observed when the nozzle section is used for steam-turbine testing.

The nozzle-discharge coefficient is the ratio of the actual flow to the theoretical flow, as calculated by the formulas to be used with this method. A calibration by the use of weighed flows should be made for each new nozzle.¹

Differential Mercury U-tubes.—In the application of this method of steam-flow determination, it is necessary to use accurate U-tube differential gages. The error of the U-tubes should be not more than 0.01 inch of mercury. To insure this accuracy, it is necessary to use slides on the mounting of the U-tube. Such a U-tube should have a plate in front of the mercury column and another plate behind the mercury column, so that the observer can look across the tops of the plates to set them exactly on the top of the meniscus of the mercury in the U-tube.

Precautions for Conduct of Steam-consumption Test.—Fluctuations of the steam pressure in the pipe where the steam flow is being measured

¹ When throat pressure holes are used for differential-pressure measurements, the nozzle-discharge coefficient is the same as that of a nozzle freely discharging into the atmosphere. For the flow formulas used for this method, the coefficient is very nearly a true friction coefficient, and its value is nearly 0.99 over a wide range of velocities. When pipe holes are used, the coefficients are slightly lower, owing to greater magnitude of the differential pressure. This is because of partial evacuation of the region back of the nozzle end due to entrainment by the nozzle jet of the surrounding fluid. In either case the coefficients derived from a calibration are reproducible with accuracy.

are objectionable, as they set up oscillations of the differential mercury columns. Steam-consumption tests of turbines should be made with the turbine governing valves blocked so that they cannot operate. Oscillations in the differential mercury columns of more than $\frac{1}{2}$ inch of mercury on either side of the average at the largest test differential pressure should be avoided. A globe valve in the steam main (preferably with a by-pass) should be used for controlling the pressure if there are likely to be large fluctuations. When, however, the steam main is open to the boilers only, the pressure fluctuations are not likely to be large. With steady mercury columns and with the pressure controlled by a globe valve, readings may be taken at 5-minute intervals in a 1-hour test; and all the readings may be taken by one observer. In a shorter test, or if the steam-main pressure varies considerably, readings must be taken oftener, and two or more observers may be needed.

When a suitably designed and carefully constructed nozzle is used, the observed differential pressures can usually be made to agree within 1 per cent, even though sufficient accuracy can be obtained if the differential pressures have a variation of as much as 2 per cent. In the case of what appears to be excessive variation in the differential pressures, it is likely to be found that the inaccuracy is due to the presence of deposits of some kind on the inner edges of some of the holes in the nozzle throat. Provision should be made in the set-up of such apparatus for the attachment of blow-off valves in the pipes for pressure measurement, so that the deposits may be removed when these valves are opened.

Formula for Small Differential Pressure $\left(\text{up to } \frac{p'}{p_1} \div 0.10 \right)$.—By the use of the symbols listed below, steam flow through an accurately constructed *steel* nozzle is given by formula (46).

w = flow, in pounds per hour.

p_1 = high-side static pressure, in pounds per square inch absolute.

p' = differential pressure, in pounds per square inch, or difference between p_1 and absolute low-side static pressure.

For most work, p'/p_1 will not exceed about 0.10.

v_1 = specific volume at high-side conditions, in cubic feet per pound (from Keenan's "Steam Tables" in the Appendix.

t_1 = high-side temperature, in degree Fahrenheit.

t_s = standard temperature at which micrometer for measuring nozzle diameter is correct, usually 68°F.

D = nozzle diameter, in inches, as average of a number of careful measurements with a micrometer which is at approximately the temperature of the nozzle.

D_1 = diameter, in inches, of pipe preceding nozzle where p_1 is measured.

n = exponent for isentropic expansion of superheated steam. Its value is 1.28

c = coefficient of discharge. With tapped throat holes this may be taken as 0.99.

Formula for Small Differential Pressure, up to $p'/p_1 = 0.10$

$$w = \frac{1,890cD^2[1 + 0.000013(t_1 - t_s)]\sqrt{p_1 p' \left(1 - 1.17 \frac{p'}{p_1}\right)}}{\sqrt{p_1 v_1} \sqrt{1 - \left(\frac{D}{D_1}\right)^4} + 1.562 \left(\frac{D}{D_1}\right)^4 p'/p_1} \quad (46)$$

Speake¹ has proposed a simplification of a more general form of the foregoing formula by using a term Y as an expansion factor, the values

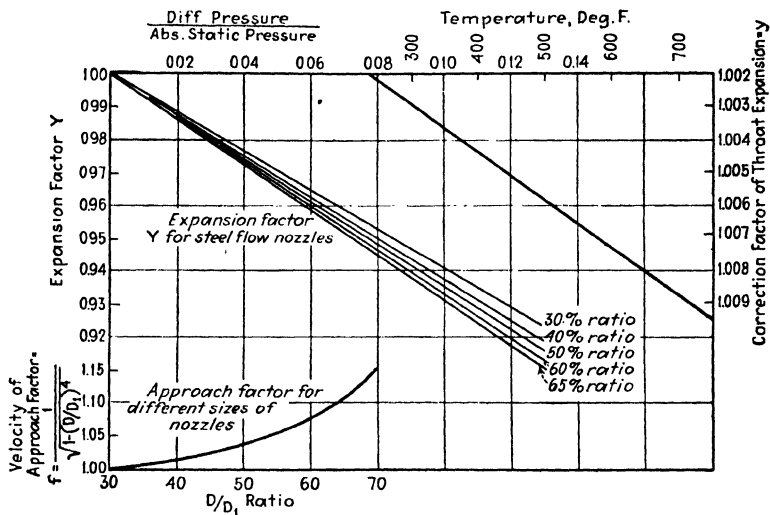


FIG. 201.—Values of nozzle expansion factor.

being given by curves in Fig. 201 for any set of conditions. In this figure, the abscissas on the upper part of the curve sheet are ratios of the differential pressure to the absolute inlet static pressure, with the ordinates at the upper left corresponding to values of the expansion factors Y for steam flow for different ratios of $\frac{D}{D_1}$ as shown by the diagonal lines. It is stated that this graphic method as applied to the formula below checks the Moss and Johnson formulas within 0.1 to 0.2 per cent. The modified formula to be used with the curves in the figure is given below:

$$w = \frac{1,890cD^2[1 + 0.000013(t_1 - t_s)]\sqrt{p'} \times \sqrt{Y}}{\sqrt{1 - \left(\frac{D}{D_1}\right)^4} \sqrt{v_1}} \quad (47)$$

¹ *Trans. A.S.M.E.*, Vol. 55, No. 11, p. 164.

By substituting the symbol f for the *approach factor* $\frac{1}{\sqrt{1 - \left(\frac{D}{D_1}\right)^4}}$,

these values can be obtained from the upper left-hand part of Fig. 201, in which the values of f for each ratio of $\frac{D}{D_1}$ are plotted.

Values of the correction factor for throat expansion

$$[1 + 0.000013(t_1 - t_2)]$$

can be plotted as shown in the right-hand part of the figure for different temperatures of the steam. The corresponding value of

$$[1 + 0.000013(t_1 - t_2)]$$

is the term y .

Thus Eq. (47) can be further simplified to

$$w = 1,890cfD^2y\sqrt{\frac{p}{v_1}}Y$$

which is suitable for rapid calculations and further can be used for any calculation of steam flow through steel nozzles regardless of the *quality* of the steam and the size of the differential pressure produced by the nozzle.

This method of measuring steam flow with nozzles has been recommended by the A.S.M.E. Research Committee on Fluid Meters¹ and has therefore satisfactory standing in professional work.

Flow of Steam Measured with Pitot Tube Meters.—One of the types of steam-flow meters made by the General Electric Company is an

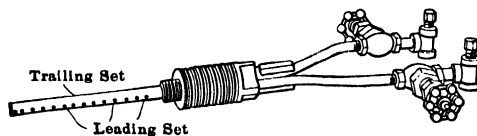


FIG. 202.—Nozzle plug for steam meter.

application of a Pitot tube of unusual design. A nozzle plug (Fig. 202) is inserted into the steam pipe and in this plug there are two sets of holes each communicating with a separate tube starting from the end of the plug. The “leading” set of holes of a nozzle plug is subjected to velocity plus static pressure, while the trailing holes are subjected to velocity less static pressure only. The principles of operation are therefore the same as for the measurement of air by the Pitot tube (see page 184). The part actuating the recorder consists of two cups connected by a hollow tube forming an elongated U-shaped vessel which is filled with mercury.

¹ *Trans. A.S.M.E.*, Vol. 55, No. 11, p. 164, 1933.

This vessel is balanced on knife edges. The cups are connected by flexible steel tubing to pipes running to the nozzle plug. In the operation of the instrument the excess of pressure in the leading holes of the nozzle plug causes the mercury to shift its level and stand higher in the cup connected to the trailing holes. As a result of this unbalancing of weights, the whole vessel will swing on the knife edges until equilibrium is established.

The Flow of Water.—When the quantity of water to be measured is not too large, it is most accurately determined by weighing in tanks placed on scales, or by direct measurement of volume in calibrated tanks

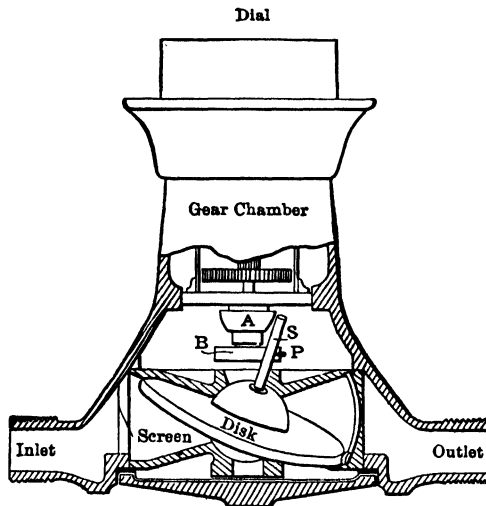


FIG. 203.—Pulsating-diaphragm water meter.

or barrels. Sometimes it is impracticable to weigh or measure the volume of the water directly, particularly when it must be measured **under pressure**. For measurements in pipes up to 2 or 3 inches in diameter, a **water meter** is generally used.

A great many types of water meters are sold commercially and not very many are accurate, so that it is absolutely necessary to calibrate them at least before and after a test, under the same conditions of temperature, pressure, and rate of flow. In many plants where meters are used constantly, suitable connections are made to the discharge from the meter, so that at any time the flow through it can be diverted into a tank in which it can be measured by volume or weighed.

One of the best types of water meters is illustrated in Fig. 203. This belongs to the class operating with a "pulsating diaphragm." The inclined shaft *S* on this diaphragm, traveling around in contact with the peg *P* on the plate *B*, moves the counting mechanism through inter-

mediate gears. This diaphragm in the **Thomson-Lambert meter** (Fig. 204) is made of hard rubber reinforced with a steel plate, which makes it much more durable than those made without reinforcing. As the side chambers are alternately filled and emptied, the diaphragm is moved up and down with a "pulsating" motion and operates the recording mechanism. The diaphragm divides the measuring chamber into two compartments of equal volume. While one of these is filling, the other

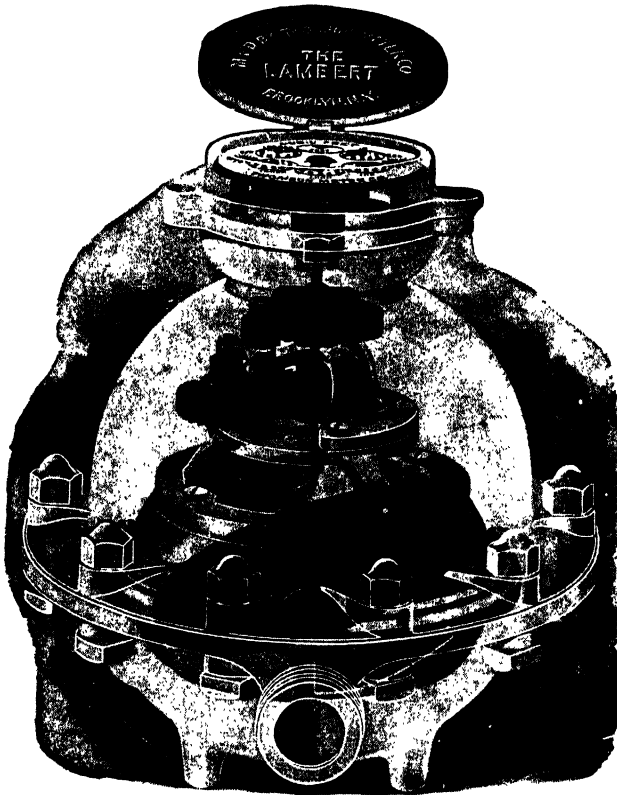


FIG. 204.—Thomson-Lambert water meter.

is emptying. For general purposes this type is probably used more than any other and for a fairly constant flow is quite accurate; but because of leakage it is not sufficiently accurate for rates of flow that at times have very low values.

Piston Water Meter.—The piston type of water meter (Fig. 205) is also used frequently. It belongs to the type operating in a cylinder by a reciprocating piston which is driven backward and forward by the pressure of the water. In this device there are two pistons side by side. Water is admitted alternately at each end by a slide valve *A* moving

on seats in the plate *S*, just above the bottom casting containing the inlet and outlet chambers. These valves are moved by contact with the inner faces of the plunger heads near the end of the travel and move them over at the proper time. The lever *L* is moved back and forth by one of the plungers to operate the counting mechanism. The cored passages in the bottom casting are too complicated to be shown clearly. The plungers at the end of their travel strike against the rubber bumpers *R*, which are provided to reduce the shock.

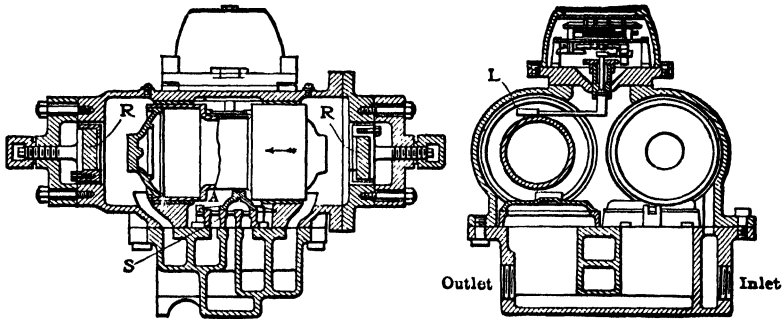


FIG. 205.—Piston water meter.

Meters actuated by the **velocity** of water are particularly suitable for measuring large quantities at low pressure. Figure 206 shows an example of this class. Water flows into the wheel *I* after entering and passing through the screen *S* as shown by the arrows. Guide vanes deflect the water horizontally and radially outward from the center into the discharge passage *A*. When the wheel revolves, it moves the counting mechanism *G* above. Objections to such meters are that they are very unreliable for small flows because of the friction of the parts, and an appreciable flow is required to start them. Friction is an important element in meters of this type, but they are not injured by moderately hot water.

The readings of a water meter are usually in cubic feet. A water meter is essentially a water motor adapted for operating the gearing connected to the counting mechanism.

Frequent calibrations of water meters are necessary because they are likely to become more or less clogged with dirt and refuse. The readings are also affected by the temperature, head, and quantity of water flowing, as well as by the amount of air carried in the water. A meter should always be calibrated at least at two or three rates of flow, as it

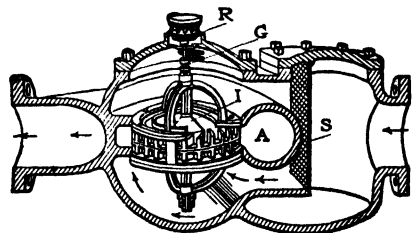


FIG. 206.—Water meter operated by velocity.

scarcely ever happens that the conditions of the test are so uniform that the meter will be used only for a certain predetermined rate of flow.¹

Willcox Water Meter.—Automatic measuring devices are often used for determining the weight of condensed steam in engine tests or the weight of feed water in boiler trials. The Willcox meter² is a most satisfactory apparatus of this kind. It consists of a tank (Fig. 207) divided by a partition *P* into two compartments *A* and *B*, one above the other. The upper compartment *A* receives the inflow of water and the lower one *B* serves for measuring. Projecting into the lower compartment is

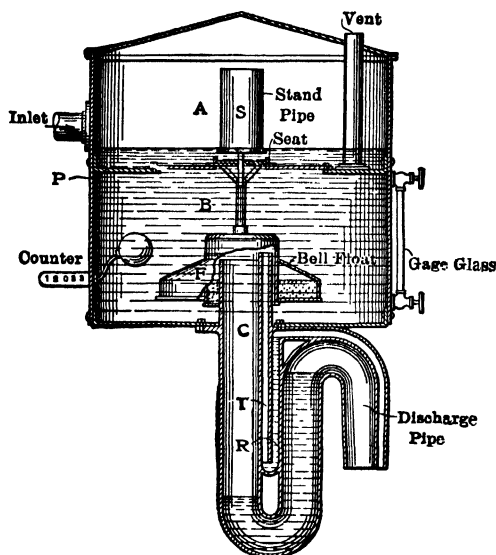


FIG. 207.—Willcox water weigher (by volume).

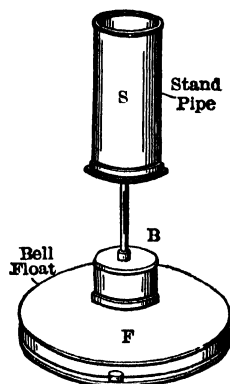


FIG. 208.—Bell float.

a U-shaped discharge pipe *C*, which is always water-sealed. The upper end of the discharge pipe is covered by a bell float *F*, which is permitted a short up-and-down movement. In the upper compartment there is a short standpipe *S*, which is simply a hollow cylinder open at the top and bottom. The bell float *F* and the standpipe *S* are connected rigidly by a vertical rod (Fig. 208) so that they move together as one piece, and this is the only moving part in the apparatus. The lower end of this standpipe has a corrugated face; and when it is down in its lowest position, its corrugated face rests on a soft seat or ring surrounding a circular opening in the partition *P*. This seat is made of a rubber composition

¹ Calibration curves are usually plotted with meter readings as abscissas and actual volumes as ordinates. A curve should be plotted for each of the several rates of flow if they are different. Curves of meter readings (abscissas) and correction factors (ordinates) are also useful.

² Willcox Engineering Co., Saginaw, Mich.

which is not injured by boiling water. The apparatus can be used, therefore, with either hot or cold water without risk.

In the operation of the apparatus, when the standpipe S is down on its seat, water entering through the side inlet accumulates in the upper compartment A until it overflows the top of the standpipe. The water then flows down through the hollow standpipe into the lower compartment until there is a sufficient amount to seal the lower edge of the bell float F . Then, as more water accumulates, the bell float rises, lifting the standpipe S from its seat, and the water in the upper compartment flows down into the lower one until the volume is that of a "unit charge" for the apparatus, when the "tripping" device discharges the water through the discharge pipe C . The "tripping" is accomplished by a "trip" pipe T , which is normally water-sealed, but which becomes unsealed when a "unit charge" has accumulated. While the water is accumulating in the lower compartment B , the water in the left-hand leg of the "trip" pipe T is being slowly pushed down because of the increasing pressure of the air under the bell float F , due to the increase of head of water, and a corresponding amount of water spills over the upper end of the right-hand leg R of the "trip" pipe into the discharge pipe. Owing to this action the water level in T is lowered until it reaches the bend in the lower end of the "trip" pipe. Under these conditions the water column in R exactly balances the head of water in the lower compartment B and the air entrapped in the float valve F has a function similar to that of a scale beam, balancing on one side the head of water in the tank and on the other side the head of the standard water column in R , which, of course, is always constant. At the instant this balance is secured, a very small amount of water added in the lower compartment and the corresponding additional spill from R will destroy this equilibrium. Then the air compressed in the bell float F and the upper part of the discharge pipe breaks the water seal in R by suddenly discharging all the water in it. When the air pressure in the bell float F is thus reduced, it drops down, carrying down with it the standpipe S in the upper compartment A . In this last operation the air is moved from the interior of the bell float F , and water flowing in to replace it will spill over the top of the discharge pipe C and will flow out of the other end until the lower chamber is emptied of the "unit charge." At this time the standpipe S becomes seated, owing to the pressure of the water above the bell float and to the downward suction of the siphon. Thus the standpipe is held tightly upon its seat only at the instant when tightness is required, that is, while the "unit charge" is being discharged. After the standpipe has seated, water again accumulates in the upper compartment A and the cycle of operations is repeated.

A mechanical counter shown at the side of the apparatus is connected to a ball float inside the lower compartment and registers the number of

times the apparatus is tripped. An automatic device of this kind is easily calibrated by weighing several "unit charges," and it can then be used with as great a degree of accuracy as can be expected with rapid weighings in tanks on platform scales. It may be expected to weigh hot or cold water with a maximum error of not more than 1 per cent for the conditions of calibration.

Leinert (Worthington) Weigher.—The apparatus shown in Fig. 209 is one of the few devices made which actually weighs the liquid to be

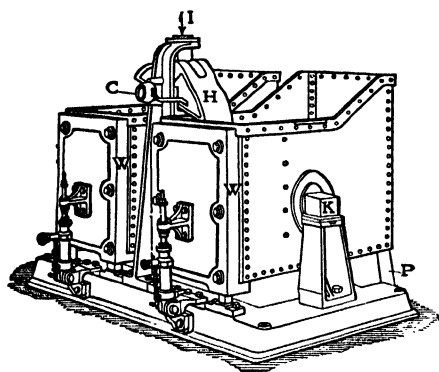


FIG. 209.—Tilting-tank water weigher.

measured. It can therefore be used for liquids of widely varying density without making corrections. It consists of a pair of open tanks supported on trunnions with knife-edge bearings (*K*) in such a way that when empty they assume the normal position as shown in the figure. After a certain quantity of liquid has entered the tank, the counterbalancing effect of the lead weights in the casing *W* is overcome and the tank tips over and the contents

are siphoned out through the pipe *P*. By this tipping action the trough *H* receiving the liquid from the supply pipe *I* is switched from the full to the empty tank, and at the same time the counter *C* is operated to register the number of times the tanks have been filled. Since weight

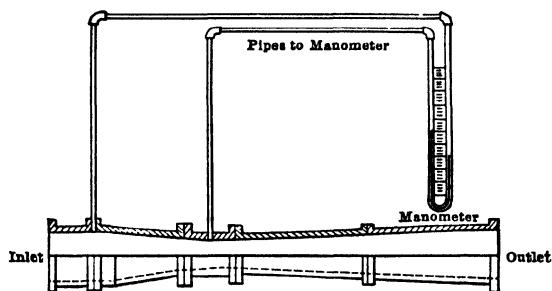


FIG. 210.—Herschel's venturi meter.

is the method of measurement, the record is independent of the temperature of the liquid; but there is always a very small variation of weight with the rate of flow. The weight per charge can be adjusted by changing the number of weights in the casing *W*.¹

¹ Henry R. Worthington, 115 Broadway, New York, and Holden & Brooke, London.

Venturi Meter.—An arrangement of piping in which there is a gradual narrowing of the section to a minimum followed by a more gradual enlargement was invented by Clemens Herschel for measuring the flow of water. This apparatus is called a venturi meter and is shown in Fig. 210. Piezometer tubes (manometers) are arranged to indicate the pressure at the sections shown. Pressures at these sections will be denoted respectively by p_m and p_n .

From Bernoulli's theorem¹ it follows that the relation between the pressure in pounds per square foot and the velocities in feet per second at the two sections v_m and v_n of a stream flowing through such a closed horizontal channel is given by

$$\frac{p_m}{d} + \frac{v_m^2}{2g} = \frac{p_n}{d} + \frac{v_n^2}{2g}, \quad (48)$$

where d is the density of the water in pounds per cubic foot. If a represents the area of a section in square feet, the volume of water flowing through any section (cubic feet per second) is

$$a_m v_m = a_n v_n = a_n \sqrt{\frac{2g(p_m - p_n)}{d \left(1 - \frac{a_n^2}{a_m^2}\right)}}. \quad (49)$$

With suitable manometers or with gages, the pressures p_m and p_n can be obtained; and since all other quantities can be represented by a constant k , we have

$$\text{Volume per unit of time} = k\sqrt{p_m - p_n}. \quad (50)$$

As usually made, the venturi tube is merely a pipe which tapers from each end toward the throat, which is usually lined with hard bronze to secure a smooth bore and has a diameter of from one-third to one-half that of the pipe line. Its total length is about eight times the diameter of the pipe. In the commercial forms of this apparatus near the inlet or upstream end and also at the throat are annular chambers encircling the tube, which communicate with the interior by numerous small vent holes. When no water is flowing in the venturi tube, the pressure will be the same in these two annular chambers; but when there is a flow of water through it, the throat pressure becomes less than the upstream pressure. The difference between the two is proportional to the square of the velocity of the water.

A recording device² has been arranged, consisting essentially of a large U-tube, serving the same function as the one in the figure, filled with mercury, supporting in each leg an iron float. These floats have toothed racks connected to their upper ends which engage with pinions

¹ See Jamieson, "Applied Mechanics," Vol. II, p. 458.

² Builder's Iron Foundry, Providence, R. I.

on the same horizontal shaft with a cam. A small wheel supporting the recording pencil rides on the perimeter of this cam, which is arranged so that the wheel rides on the greatest eccentricity of the cam and consequently the recording pencil will indicate on the chart the greatest flow when the rack is at its maximum height. The pipes connecting the U-tube with the venturi meter should always be full of water. Air cocks are provided for removing air that may accumulate in pockets.¹ Since the flow is proportional to the square root of the difference of pressure, a complicated cam device is required to permit the charts of the recorder to be made with equal divisions.

The General Electric flow meter (see page 204) is also sometimes applied for measuring the flow of water in pipes.

The Vaughan flow meter² (Fig. 211) is a mechanical device for indicating the flow of liquids in pipes. It consists of a cylindrical chamber with suitable pipe connections enclosing a slotted tube through which the liquid must pass. Enclosed in this tube

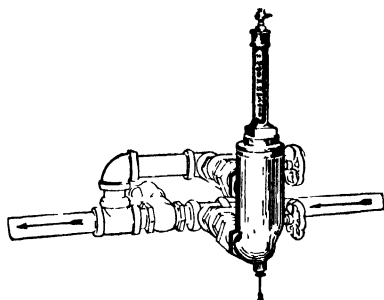


FIG. 211 — Vaughan flow meter.

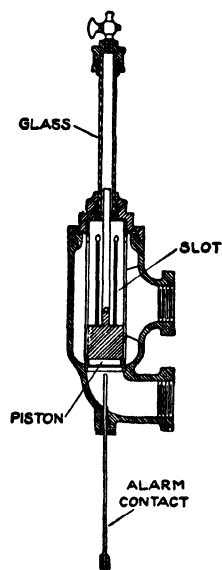


FIG. 212.—Section of Vaughan flow meter.

is a piston float which rises with increased flow of the liquid and is carried up until the exposed area of the slot is sufficient to allow the flow of the liquid through it without further rise in the level of the liquid. A scale at the side of the piston index indicates the rate of flow. There is also an electric alarm at the bottom which operates when the flow of liquid is too low. This instrument is shown in section in Fig. 212.

Flow of Water through Orifices and Nozzles.—Theoretically, the velocity of flowing water under any pressure is the same as the velocity attained by a body falling freely through a distance equal to that head h as in Fig. 213. Furthermore, this statement would be the same even if the water had no free surface, provided, however, the **pressure** at

¹ For more detailed discussion and tests see Herschel's papers in *Trans. A.S.C.E.*, November, 1887, and January, 1888; also *Power*, Jan. 23, 1912.

² Made by Gurley, Troy, N. Y.

the orifice was that due to a head h . If then there is no loss of head due to friction and eddies formed by the water passing through the orifice, the velocity of discharge, v in feet per second, is

$$v = \sqrt{2gh}, \quad (51)$$

where g^* is the acceleration due to gravity and h is the head over the center of the orifice in feet.

If a is the area of the cross section of the orifice in **square feet**, q is the quantity or volume of water discharged in **cubic feet per second**,

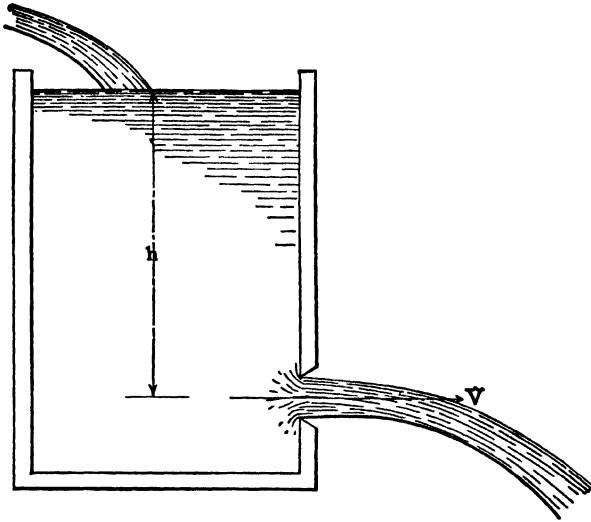


FIG. 213.— Discharge of water from an orifice.

and if it is assumed the stream is of the same cross-sectional area as the orifice, then

$$q = a\sqrt{2gh}. \quad (52)$$

Since the actual flow is less than the theoretical in most cases, and considerably less when the discharge is from a hole with sharp edges in a thin plate, a more general form may be written by inserting a suitable coefficient¹ of discharge k , then

$$q = ka\sqrt{2gh}. \quad (53)$$

For an orifice located in the side or bottom of a tank, consisting of a circular opening in a thin metal plate with a smooth **sharp** edge, the

* The value of g is approximately 32.2, so that Eq. (51) can be simplified into $v = 8.02\sqrt{h}$.

¹ This coefficient is often called the coefficient of discharge and is the product of the coefficients of contraction and of velocity.

value of the coefficient k may be taken as 0.6 for all practical purposes (see Report Power Test Committee, *Journal A.S.M.E.*, November, 1912, page 1829).

Calibration of Orifices and Nozzles.—Water under a constant pressure is often measured by observations of the flow through either orifices or short nozzles which have been carefully calibrated. The apparatus required for this calibration consists usually of a suitably arranged stand-pipe to which the orifice or nozzle can be attached so that a given head of water can be maintained¹ and barrels on scales (or tanks calibrated for volumes) are used to receive the water discharged.

A pressure of 1 pound per square inch is equivalent to a head of water at 62°F. of 27.72 inches, or 2.31 feet. A normal atmospheric pressure (14.7 pounds per square inch) is therefore equivalent to a head of 33.96 feet of water. Then for a given pressure or head the quantity of water discharged in a given time is readily obtained and the coefficient of discharge can be computed by substituting the values of quantity of discharge q , the head h , and the area a , in Eq. (53).

Data and results should be tabulated in the form given below. The relative roughness of the edge of the orifice or of the inside surface of the nozzle should be recorded.

FLOW OF WATER

Flow of water through.....
 Date.....Observers.....
 Form of orifice or nozzle.....Formula.....Diameter, feet.....
 (Sketch).....Area, square feet

No. of reading	Head in feet	Time in seconds	Total pounds or cubic feet	Pounds per second	Cubic feet per second	Coefficient of discharge k	Remarks
Average							

Curves.—Curves should be plotted for each orifice or nozzle with head in feet for abscissas and (1) the discharge (cubic feet per second) and (2) the coefficient of discharge for ordinates.

Flow of Water over Weirs.—When large quantities of water are to be measured, then orifices are unsuitable and it is customary to pass the

¹ In many places a suitable pressure tank is not available, and in such cases the calibration can be made by attaching the orifices or nozzles to pipes carrying water under pressure. The readings of the pressure gage can be reduced to the equivalent head in feet, to which must be added, if the center of the gage is higher than the orifice or nozzle, the distance in feet from the center of the nozzle to the center of the gage.

whole body of water over a **weir or gage notch**. This consists of a board placed across the stream so that all the water must pass over it. The length of the notch is usually made less than the width of the stream to give definite conditions. This is accomplished most easily by sawing the notch out of a long board and beveling the edges.

A typical arrangement for measuring the head of water on a weir is illustrated in Fig. 214. The head must be determined with great accuracy, and this is done usually by means of a **hook gage** (Fig. 215) and a suitable machinist's or carpenter's level.

The hook gage consists of a sharp-pointed hook *H*, attached to a vernier scale *V*, intended to measure very accurately the amount the hook is moved. Before taking an observation, the hook must first be submerged and then raised slowly till the point just breaks the surface of the water. The correct height of the surface is obtained at the instant when the point of the hook pierces it. The head *h* of the water flowing over the weir (Fig. 214) is obtained by setting by means of a straightedge *SE* and the level *L* the point of the hook at the same level as the crest of the weir.

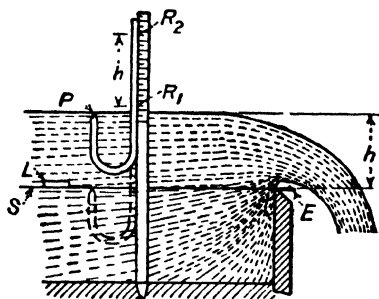


FIG. 214.—Typical weir for measuring water.



FIG. 215.—A hook gage.

The height observed in this position is called the **zero head**. It is to be subtracted from all other readings to get the head of water flowing. The hook gage must be placed in a position on the upstream side of the weir where the surface has no appreciable velocity and where there is very little disturbance due to eddies. In terms of the following symbols,

- q* = quantity or volume of water discharged, in cubic feet per second.
- h* = head, in feet, on weir measured in still water.
- b* = breadth of the weir, in feet.
- n* = number of contractions.
- k* = coefficient of discharge.

$$q = \frac{2}{3}kh^3(b - 0.1nh)\sqrt{2g}. \quad (54)$$

This is the well-known Francis formula for a rectangular notch. The

ordinary rectangular notch has two contractions, one at each side of the crest.

Where there is a relatively deep channel to make the velocity of approach small, a **rectangular weir without end contractions** is the best type.¹ End contractions are likely to be sources of error and should be avoided when possible.

Triangular notches in weirs are sometimes used. One of these in the form of a right-angled isosceles triangle is shown in Fig. 216. It has the advantage of giving the same form of stream whatever the size of the

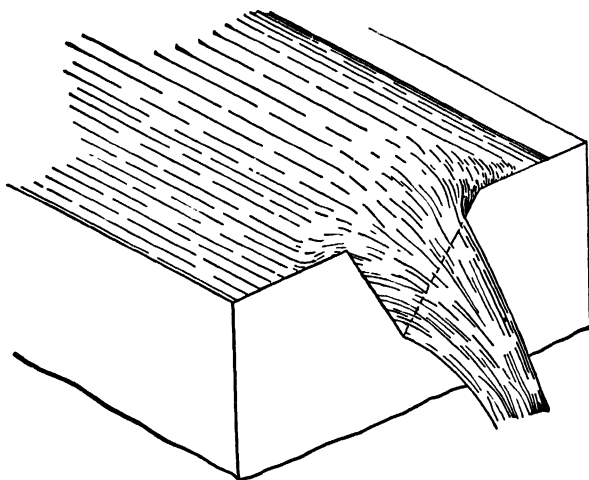


FIG. 216. Weir with a triangular notch.

notch or the height of water passing through. It is, therefore, particularly suitable for measuring a flow of water which is somewhat variable. The quantity of water discharged over a triangular weir or notch is

$$q = \frac{4}{15} k b h^3 \sqrt{2g}. \quad (55)$$

When the angle is 90 degrees,

$$b = 2h \text{ and } q = 4.26 k h^{\frac{5}{2}}. \quad (56)$$

Also when the angle is 60 degrees,

$$b = 2h \tan 30^\circ \text{ and } q = 2.47 k h^{\frac{5}{2}}. \quad (57)$$

Weir Meters.—A weir or notch will measure any quantity of water if made of a suitable size. Rectangular weirs are generally used for large quantities and notches of various shapes for small quantities. For accurate work the head on the crest is measured with a hook gage, but in many cases a float is used. The instrument used for measuring the head should not be less than 2 feet from the crest and preferably

¹ DAUGHERTY, R. L. "Hydraulics," 2d ed., pp. 82-89.

farther away. The distance of the crest from the bottom of the weir tank should be not less than three times the average head. For a weir with two contractions, the width of the tank should be not less than three times the width of the weir. To maintain the surface free from ripples, baffle plates, preferably of perforated sheet metal, should be located between the supply pipe and the instrument for measuring the head.

Lea's recorder for V-notch weirs is successfully used in many services for measuring water. Water level is measured by a sheet-metal float usually about 12 inches in diameter which is attached to the vertical shaft *S* of the recorder in Fig. 217. On the upper end of this shaft is a rack *R* meshing with a pinion on the left-hand end of the horizontal shaft carrying the drum *D* which has on its curved surface a spiral band over which a trailing arm *F* is fitted. The curvature of the spiral band has been made to conform to a logarithmic curve, so that increments of movement of the trailing arm

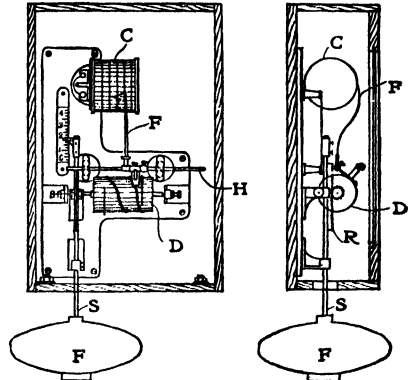


FIG. 217.—Lea's recorder.

are proportional to the quantity of water flowing, and not to the up-and-down movement of the float. The trailer *F* is connected to a pen arm making a record on the paper drum *C*. These charts will therefore have equally spaced ordinates and the area under the curve traced on them is proportional to the quantity of water flowing in a given time.

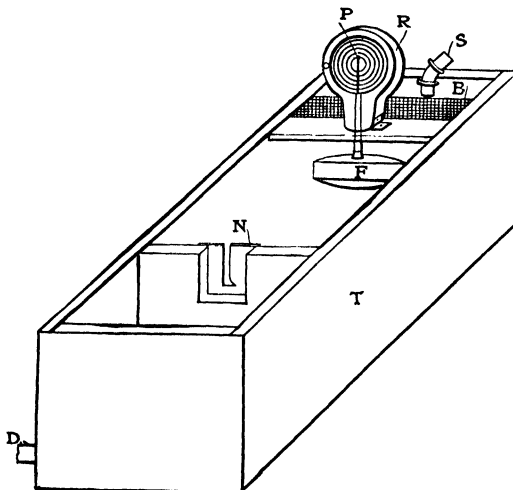


FIG. 218.—Moyer's recording weir.

discharges through the supply pipe *S*. A float *F* is located under the recorder *R*. The vertical shaft of the float, held in line by small ball-bearing guides, is connected to the pen point *P* directly, without any intervening gears or linkages. Baffle plates *B* are placed between the

Moyer's recorder (Fig. 218) consists of a weir tank *T* into which water dis-

supply pipe and the float to eliminate ripples and steady the float. The notch *N* follows almost exactly the theoretical lines for a flow proportional to the head, accurate allowances being made also for end contractions. Radial ordinates traced by the pen point *P* are therefore proportional to the rate of flow. By measuring these charts with a Bristol-Durland averager for circular charts (see page 112) the quantity of water flowing in a given time can be accurately obtained.

The charts are graduated in pounds of water per hour instead of cubic feet. This is made possible by the automatic temperature correction of this apparatus. For a given weight of flow as the temperature increases or decreases, the head increases or decreases correspondingly and *vice versa*; but at the same time the displacement of the float increases as the water becomes lighter by reason of being hotter. These two influences therefore counterbalance, making the weight discharged in a given time proportional to the ordinates of the curve traced by the pen of the recorder. This apparatus is accurate to 0.5 per cent for temperature correction.

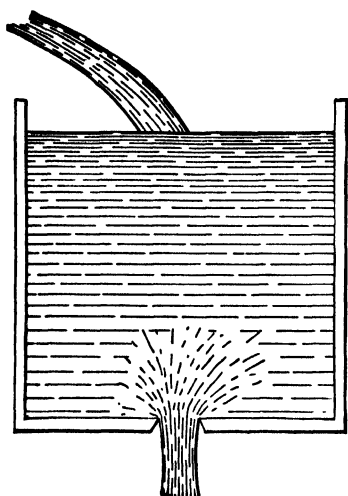


FIG. 219.—Best kind of orifice for engine tests.

than with an orifice. Where great accuracy is desired and the quantity of water to be handled is not too large, an orifice calibrated and used in the bottom of the tank as shown in Fig. 219 is to be preferred to measurements with a weir. This remark is particularly applicable in connection with the measurements of cooling (circulating) water in tests of large steam engines and turbines.

Calibration Data Sheets.—Use the same form for data as given for calibration of orifices or nozzles on page 214.

Curves should be plotted with heads for abscissas and (1) the discharge (cubic feet per second) and (2) the coefficient of discharge for ordinates.

Weighing Liquids in Tanks.—In order to weigh liquids under a continuous flow, two tanks, usually made of sheet metal, are generally used. Each tank is placed on a scale and the liquid is alternately discharged into each. The discharge pipe is usually arranged so that it is

movable from one tank to the other by turning on a screw thread. At the side of each tank near the bottom a so-called "quick-opening" valve or cock is provided for rapidly emptying the tanks. These discharge valves should be large, because the more rapidly the tanks can be emptied the greater the quantity of water the arrangement will handle.¹

When only one platform scales is available, an arrangement like that shown in Fig. 220 can be used efficiently. The larger tank is placed on the scales and the smaller one is supported on a platform or bench at such a height that it can be readily discharged into the larger tank. During the operation of weighing and emptying the larger tank, the liquid is discharging into the smaller one; and when the discharge is again directed into the larger tank, the valve or cock on the smaller one

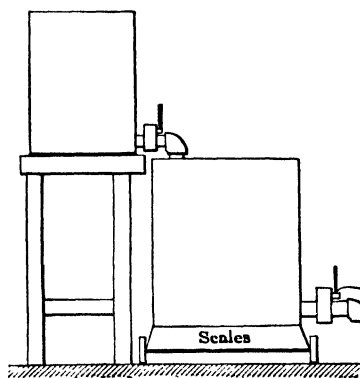


FIG. 220. Weighing device.

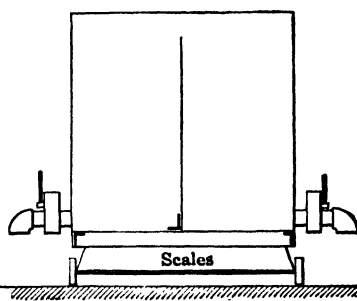


FIG. 221.—"Double" tank.

is opened so that its contents will be included in the next weighing of the larger tank.

For weighing feed water in boiler tests the reverse of this arrangement is frequently applied. There are as before two tanks or barrels; the more elevated one is on a platform scales, and the attendant doing the weighing empties weighed quantities of water into the lower tank as needed to supply the feed pump. In the lower tank the water level must be the same level at the end as at the beginning of a test.²

Where the flow is absolutely constant, as in the discharge of water from nozzles or orifices with a constant head, a tank may be filled at intervals, observing accurately with a stop watch the time for filling and weighing each time. The average of several such determinations gives a fairly accurate result.

¹ The discharge can be increased by attaching a short pipe to the discharge side of the valve, which by reducing the contraction increases the flow.

² For determining these levels in the tank a water-gage glass is very convenient. If, however, there is no gage glass on the tank, marks can be made with a knife or by painting a line on the inside of the tank.

Liquids are also often measured instead of weighed in calibrated tanks. In every case the temperature of the liquid must then be observed. In some cases the tanks have graduated scales at the side of a glass water gage from which the volume of water can be observed; or again there is only a single mark up to which the tank is to be filled each time. Establishing the exact level for a large surface is not an easy matter and to make this method more accurate the marks up to which the tank is to be filled are preferably put on a portion of the tank at the top which has been made considerably smaller in size than the rest of the tank.

It is a very poor method to fill tanks up to the rim on account of the variableness of the meniscus, which may vary from various causes.

Actual weights corresponding to measurements of volume are always varying with the temperature of both the liquid and the tank or collecting vessel. It is therefore necessary in every case to determine by actual tests the weight corresponding to the volume of a tank for a given liquid at various temperatures and apply a calibration curve for temperature variations to all measurements. Calibrations should be made also with the inside wetted surface in as nearly the same condition as it will be after each emptying in a test.

An ingenious method of measuring the flow of a liquid is illustrated in Fig. 221. It shows a tank separated by a partition into two parts. The partition is not quite so high as the sides of the tank. In operation, one of the halves is filled up to the top of the partition, permitting any excess to flow over into the other half. Then the supply pipe is swung over to the second half which is filling while the first half is being emptied. The tanks must be emptied, of course, more rapidly than they are filled.

In order to make easier the supervision and checking of tests, it is desirable that the quantities weighed or measured should be as nearly equal as possible.

Automatic scales provided with a continuously operating controlling device have been successfully developed. The poise on the weighing beam is set in motion when the article to be weighed is put on the scales; and when the beam shows it is balanced, the poise stops automatically. The weight is usually registered by means of a counting device, but a printing recorder can also be used.

PERMISSIBLE VELOCITY AND ACCURACY OF DEVICES FOR THE MEASUREMENT OF FLUIDS

	Measure- ment	Permissible velocity, feet per second	Probable error, per cent
Gas meters:			
"Wet" meter.....	Volume	100 r.p.m.	$\frac{1}{2}$
"Dry" meter.....	Volume		5-10
Pitot tube.....	Velocity	Above 8	1
Anemometer.....	Velocity	1-40	1-5
Electric meter.....	Weight	3-500	$\frac{1}{2}$
Receiver.....	Volume	Unlimited	1
Orifice.....	Weight	Unlimited	2
Venturi tube.....	Velocity	Unlimited	1
Steam meters:			
Pitot tube.....	Velocity	Above 10	1
Orifice.....	Weight	Unlimited	$\frac{1}{2}$
Float.....	Volume	50-200	2
Water meters:			
Impulse wheel.....	Velocity	5-100	1
Rotary disk.....	Volume	100 r.p.m.	1
Piston.....	Volume	150 strokes per min.	1-5
Rectangular weir.....	Volume	Unlimited	$\frac{1}{2}$ -1
V-shaped weir.....	Volume	Unlimited	$\frac{1}{2}$ -1
Flow-proportional-to-head weir.....	Volume	Unlimited	$\frac{1}{2}$ -1
Automatic tank.....	Volume	Up to 20 lb.	1
Venturi tube.....	Velocity	Varying	$\frac{1}{2}$
Pitot tube.....	Velocity	10-400	1

CHAPTER VIII

CALORIFIC VALUE OF FUELS—SOLID, LIQUID, AND GAS

"Calorific power" is a term applied to the quantity of heat generated by the complete combustion of a definite quantity of fuel. In order to insure rapid and complete combustion, the fuel is **preferably** burned in an atmosphere of oxygen under pressure. This calorific power of fuels is expressed in the English system as **British thermal units per pound**, and in the metric system as **calories per gram**.

In fuel calorimetry it is always assumed in engineering calculations that at about the usual "room" temperatures the specific heat of water is constant, so that the weight of reasonably pure or distilled water in pounds times the change in temperature in degrees Fahrenheit is the heat change in British thermal units (B.t.u.). Similarly, the weight of water in kilograms times the change of temperature in Centigrade degrees is the heat change in kilogram-calories (French units).¹ For conversion constants see page 74.

The quantity of heat generated by combustion is measured by the rise in temperature of a given weight of water in a calorimeter of which the **heat equivalent** or **water equivalent** k has been determined, and the temperature of any gas escaping has been reduced to that of the room.

Let w_f = weight of the fuel, in grams.

w_w = weight of the water, in grams.

k = heat or water equivalent² of the calorimeter, in grams.

t_1 = initial temperature of water, in degrees Centigrade.

t_2 = final temperature of water, in degrees Centigrade.

Q = total heat generated, in calories.

¹ In determinations of the calorific value of *solid and liquid* fuels it is customary to measure temperatures in Centigrade degrees and weights in grams (page 224).

² Heat equivalent is used to express the heat-absorbing effect of the calorimeter as equivalent to that of a weight of water. This may be found (1) as for calorimeters used for determining the quality of steam by the hot-water method (see p. 95); (2) by taking the sum of the products of the weights and specific heats of the various parts of the calorimeter (see "Calorific Power of Fuels," by H. Poole, pp. 14, 15); or (3) (method recommended by Power Test Committee of A.S.M.E.) by comparing the calorific results obtained with those that should have been secured, if there had been no absorption of heat in the calorimeter itself, by the combustion in oxygen gas of some substance of which the heat value is known, as, for example, pure sucrose, carbon, naphthalene, or benzoic acid. Samples of the materials of which the heat value has been worked out accurately are available at small expense from the U. S. Bureau of Standards. The standard sample should be made into a pellet (see p. 229)

Then the calorific value H per gram of fuel in calories is

$$H = \frac{Q}{w_f} = \frac{(w_w + k)(t_2 - t_1)}{w_f} \quad (58)$$

Heat of Combustion of Solid and Liquid Fuels.—At the present time, there is a general agreement among testing experts that the heat values of solid and liquid fuels can be accurately determined with the bomb-type calorimeter (page 229). Among those that have received favorable mention by the A.S.M.E. are the following: Mahler, Atwater, and Emerson. Bombs for heat of combustion determinations should be made of acid-proof alloys or of solid steel that is lined with gold or platinum. Bombs made of phosphor bronze, and which are gold-plated on the inside, have also given satisfaction. Formerly, many of the steel bombs used for this service were lined with porcelain, but the disadvantage of this kind of lining is that it is likely to crack and thereafter will deteriorate very rapidly, while gold and platinum linings are almost indestructible for this kind of use. The valves that are used for admitting oxygen to the bombs and the insulated electrodes that are used for making the electric spark that produces combustion are usually the first parts of a bomb calorimeter to fail and should, therefore, be designed so that they can be easily replaced. In some designs the screw threads that are used to attach the cap to the bomb have such small threads that they wear out

weighing about 1.5 grams which immediately after weighing should be put into the calorimeter and burned in commercially pure oxygen gas at about 400 pounds per square inch. After correction for radiation (see p. 233) the discrepancy in the heat balance is the product of the required heat or water equivalent and the temperature range. The determination of the heat equivalent of the calorimeter should be based on the average value found by at least five combustions of standard sucrose, benzoic acid, or naphthalene in the calorimeter, the same procedure being followed in the combustion of these standard materials as in the combustion of coal and other fuel samples. Standard materials for combustion may be obtained with certificate of heating value and corrections, if any, from the U. S. Bureau of Standards, Washington, D. C. These are the only combustion standards accepted in connection with the Test Codes. The heat or water equivalent of the calorimeter is given by¹:

$$\frac{(\text{Weight of standard fuel} \times \text{its certified heating value}) + (\text{fuse-wire correction})}{\text{Corrected temperature rise}}$$

Detailed directions for very accurate determinations are given in *Circ. 11*, U. S. Bur. Standards.

¹ It will be noted that the correction for fuse wire in this equation is additive, whereas it is subtractive in the actual determination of heating value of a fuel. Burning of the fuse wire actually develops a known amount of heat in the calorimeter which goes to increase the temperature rise. In the case of standardization, the heat or water equivalent of the calorimeter is obviously equal to the total (known) heat developed therein divided by the temperature rise. In burning a fuel in the calorimeter, the total heat developed is the heating value of the fuel plus that due to the fuse wire. The correction must therefore be subtracted from the total in order to obtain the true heating value of the fuel.

quickly. On this account, a comparatively large pitch for the screw threads should be used.

Stirrers for circulating the water in which the bomb calorimeter is immersed should preferably be multibladed and operated at less than 200 revolutions per minute. Such stirrers operated at a relatively low speed will not develop sufficient heat from their running to introduce an error in the tests. On the other hand, if a single small propeller is used for stirring and is operated at a very high speed, the heat development from the stirring may be excessive and large enough in amount to require a correction in the test results.

Thermometers for Use with Bomb Calorimeters.—For accurate determinations of the heat of combustion of solids and liquids with a bomb calorimeter, an especially good temperature-measuring instrument should be used. Platinum resistance thermometers are recommended by the A.S.M.E. for this service because of the accuracy that is easily obtainable with such an instrument. The temperature measurement should be readable by interpolation to $0.001^{\circ}\text{C}.$,¹ and, of course, the instrument should be very carefully calibrated, either by direct test or by the comparison method (page 51). Good mercury-in-glass thermometers when carefully calibrated are, however, sufficiently accurate for this kind of testing.

Heat Equivalent of Calorimeter.—The calorimeter determination of the heat equivalent or water equivalent of a bomb calorimeter should be determined according to the test codes of the A.S.M.E. (page 233) by duplicate determinations of the heat of combustion of samples of benzoic acid or naphthalene, as obtained from the U. S. Bureau of Standards. Heat determinations made with such samples should agree within 0.3 per cent for the same observer, and 0.5 per cent for different observers.

The **calorimeter capacity** or heat-equivalent value of the calorimeter will be difficult to obtain accurately if it contains a large amount of material of low-heat conductivity, for the reason that low heat-conductivity materials introduce a time lag into the results. On this account, for the best accuracy in determinations, the calorimeter and its auxiliary parts should be made as much as possible from materials of high- rather than low-heat conductivity.

The evaporation of water may also introduce an element of inaccuracy; and in order to avoid evaporation, accurate determinations of heat of combustion of fuels are made with an oil seal over the water surrounding the calorimeter.

Adiabatic Principle as Applied in Temperature-measuring Instruments.—Some fuel calorimeters as, for example, the Emerson bomb type, are provided with a so-called "adiabatic" jacket around the water con-

¹ In fuel calorimetry it is customary for American engineers to use the Centigrade scale of temperature.

tainer. Such a calorimeter jacket is intended to be filled with a liquid, usually water, and has immersed in its liquid an electric heater with rheostat adjustments so that the water in the jacket can be raised in temperature, so as to maintain in the calorimeter and in the jacket always the same temperature. This adiabatic jacket is, therefore, a device to reduce to a negligible amount the radiation loss from a fuel calorimeter. Another method of making the radiation loss very small is to surround the water container with a vacuum-walled jacket which will serve as a non-conductor of heat in the same way that the ordinary vacuum bottle does. It is claimed that a calorimeter equipped with a vacuum-walled jacket will have so little radiation loss that, if the temperature of the calorimeter water is close to that of the room (and, of course, also close to that of the jacket walls) at the beginning of the calorimeter tests, no radiation correction need be applied.

Precautions for Accuracy in Fuel Calorimeter.—A small room, preferably one without outside exposure, is much better suited for fuel-calorimeter work than a large open laboratory room. If it is necessary to do work requiring the use of a fuel calorimeter in a large open laboratory, it is advisable to construct a temporary partition at least 10 feet high to prevent the direct contact of air currents with the apparatus. It has been found that the best results are obtained with fuel calorimeters by keeping room conditions at such values that all determinations are made with practically the same temperature range. Where high-accuracy resistance thermometers are used in calorimeter work, care should be taken to protect them from laboratory fumes, since even slight corrosion of the electrical contacts may introduce measurable errors. It is a wise precaution also to set up the calorimeter in such a way that it will always occupy the same position relative to the auxiliary equipment that is provided in the jacket. Contact of a part of the calorimeter with large amounts of insulating material which adds to the heat equivalent of the instrument is to be avoided, since this material will necessarily add to the heat capacity of the instrument and will therefore have its effect in increasing the temperature-time lag. Excessive temperature-time lag is objectionable, since it tends to render the combustion intervals indefinite.¹

In fuel calorimetry work where mercury-in-glass thermometers are used instead of electric-resistance thermometers, as first recommended, it is usually desirable to take temperature readings of the water surrounding a bomb calorimeter with the aid of a small reading telescope of the kind

¹ The A.S.M.E. committee that prepared the Power Test Code on Heat of Combustion recommends that, where it is necessary to make as many as 50 determinations of the heat of combustion of fuel in a day, it is economically possible to use fuel calorimeters that are provided with complete water jackets controlled by thermostats. This provision will eliminate the taking of observations for determining the cooling correction (radiation) (p. 233).

frequently furnished with calorimeter thermometers (magnifying power usually 10 diameters). With the aid of such a telescope, it is possible to read 0.01°C. with considerable accuracy. Small reading telescopes of the kind that are fastened to the thermometer stem by a spring clip may be satisfactorily used if the calorimeter is protected from temperature changes due to the proximity to the water jacket of the person taking the readings. It is a better practice, however, to use a reading telescope that can be mounted at some distance from the calorimeter so as to slide up and down a vertical rod with a suitable counterpoise provided so that it will be in a balanced position at any point on the thermometer scale and may be easily shifted up or down to follow changes in the temperature indicated by the thermometer.

Mercury-in-glass thermometers should always be tapped before a reading is taken to eliminate errors due to sticking of the mercury column in its capillary tube. It is not uncommon to find that, without this tapping of the thermometer, there is a lagging error as much as 0.01°C. The tapping of the thermometer is usually done by a light rod or a pencil, but it is sometimes more convenient to use an electric buzzer with a rubber-tipped armature that strikes the thermometer stem very lightly. Where a resistance thermometer is used, on the other hand, the electrical indicating instrument that is used for the readings must similarly be tapped to avoid the possibility of errors due to sticking of the moving parts. For all official tests of the heat of combustion of fuels, the Test Codes of the A.S.M.E. require that the thermometers used "shall be calibrated for this particular purpose by the U. S. Bureau of Standards or by another physical testing laboratory of high standing."

The Beckmann differential mercury-in-glass thermometers (page 72) that have an adjustable range of from 5 to 6°C. are especially well suited for fuel-calorimetry work. Such thermometers should be graduated to 0.01 or 0.02°C. intervals, so that they may be easily read to 0.01°C. , preferably by means of a reading telescope and a cathetometer.¹

Typical Bomb Calorimeters.—Formerly the calorimeters used for burning fuels in an oxygen atmosphere were arranged for combustion at **constant pressure**; but since it was found that more reliable results could be obtained generally with apparatus maintaining a **constant volume**, the former type is not now much used. When the combustion takes place at **constant volume**, the vessel receiving the charge of fuel and oxygen

¹ A *cathetometer* consists of a vertical shaft supported on a tripod. A sliding carriage is provided to move up and down on a carefully fitted vertical track. The sliding carriage carries a telescope and can be clamped in any position on the track corresponding to the elevation at which the telescope is to be used. Minute movements of the telescope can then be obtained by adjusting a vernier which is a part of the carriage. The moving parts are supported on cords which pass over pulleys at the top of the supporting standard and have counterbalancing weights on the opposite side of the pulleys.

must be designed to withstand a great pressure, and therefore on account of the massive construction required the vessel is called a **bomb calorimeter**. The essential part of such a calorimeter is the strong steel vessel or bomb similar to Fig. 222. It consists of a steel shell *S* having a capacity of about 50 cubic inches and capable of resisting with safety a pressure of about 4,000 pounds per square inch. This shell is usually provided with a coat of enamel or a lining of platinum, gold, or nickel on the inside and is nickel-plated on the outside. The coating or lining on the inside is intended to resist corrosion and oxidizing action during the combustion. The advantage of the nickel lining over the coat of enamel is that when it is worn out or broken it can readily be replaced and at much less expense than the enamel. The shell is closed at the top by an iron cover or cap which is to be made tight by screwing down on a lead washer with considerable force, using a long wrench. At the top of this cover or cap there is a conical seated valve, which is screwed in through the gland and stuffing box *G*, by attaching a wrench at *P*. The valve and its seat are made of good nickel, as this metal is not easily oxidized. A wire electrode *B*, which is well insulated from the cover, extends into the shell and conducts the electric current for firing the charge of fuel, which is placed on a platinum dish or crucible supported by another wire *A*, attached to the cover on the inside.

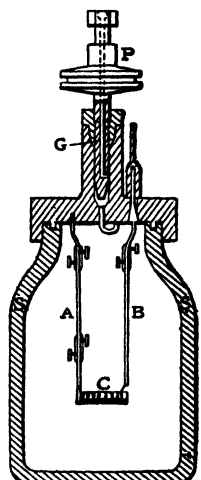


FIG. 222.—Section of bomb calorimeter.

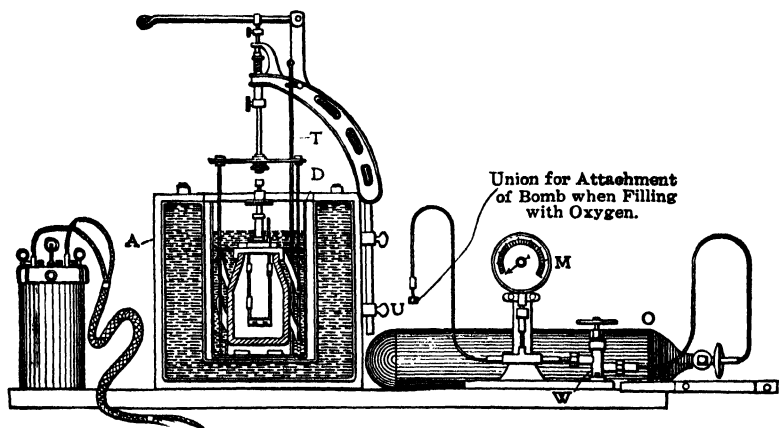


FIG. 223.—Complete Mahler apparatus.

The complete Mahler apparatus is shown in Fig. 223, showing the cylinder of oxygen *O*, the pressure gage *M*, and the calorimeter vessel *D*.

The end of the conical-seated valve (Fig. 222) is attached by means of pipe connections, preferably flexible, to the union *U* and to the valve *W*, which because of the high pressure should be opened slowly and

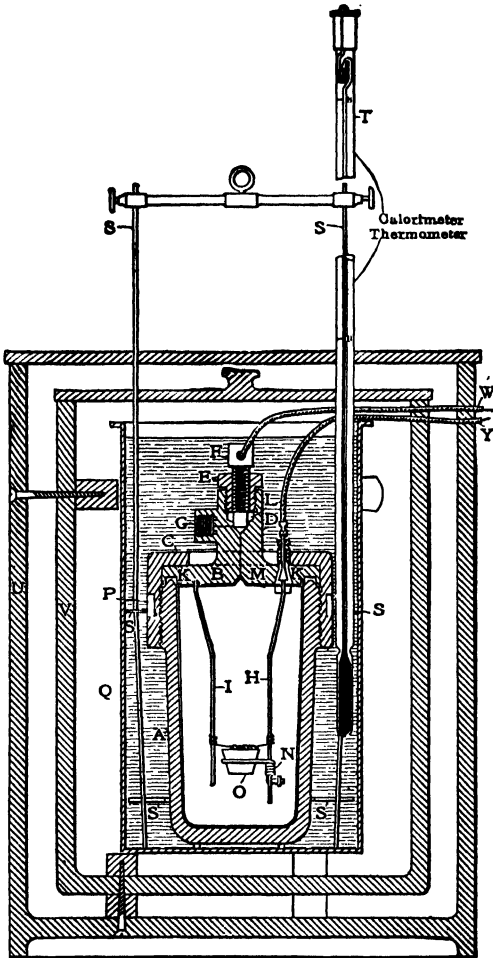


FIG. 224.—Atwater's fuel calorimeter.

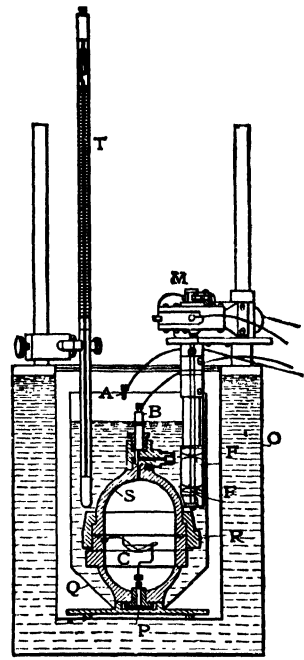


FIG. 225.—Emerson's fuel calorimeter.

carefully, and sufficient oxygen allowed to pass into the bomb to provide a considerable excess above that actually required. The pipes for connecting the bomb to the oxygen cylinder should connect also with a pressure gage as shown, so that the pressure in the bomb can be regulated. For the combustion of coal a sufficient volume of oxygen is admitted to Mahler bombs of the usual size to make the pressure in the bomb from 350 to 375 pounds per square inch.

"The Mahler type of calorimeter is recognized as the most complete and accurate apparatus of its kind. Where the engineer does not have this instrument or some other reliable calorimeter of similar construction, the heat units can be determined by sending samples to a testing laboratory where such instruments are used." (Report of Power Test Committee of A.S.M.E., *Mech. Eng.*, November, 1912, p. 1698.)

A modification of the Mahler bomb calorimeter has been designed by Atwater¹ (Fig. 224) and another by Emerson (Fig. 225). The former consists of the shell of the bomb *A*, the cap *C* screwed on numerous threads to the shell and holding down the cover *B*. Into the vertical

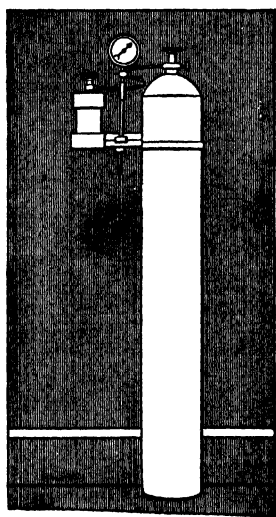


FIG. 226.—Apparatus for charging a bomb calorimeter with oxygen.

neck of this cover a screw *E*, holding another screw *F*, is fitted and is to be turned down tightly, a lead washer serving as "packing." A small passage for the admission of oxygen from *G* is opened and closed as required by turning the screw *F* operating a needle valve. A wire *H* of platinum or other nonoxidizable metal passes through the cov-

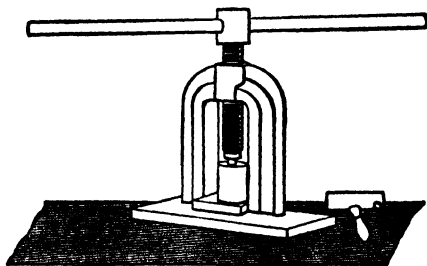


FIG. 227.—A pellet press for compressing samples of fuel.

er *B* and is insulated from it by a collar of hard rubber. Another wire rod *I* is attached to the lower side of the cover and electrical connection is made between the two wires *H* and *I* by a small iron wire stretched between them. A platinum crucible provided for receiving the fuel is supported by a "screw" ring. Ball bearings of hard steel are sometimes placed between the cover and the cap to reduce friction when screwing down. Holes located in the sides of the cap are for attachment of a long spanner wrench when turning down the cap. A hand-stirring device *S* is used for agitating the water in the vessel *Q*.

The usual arrangement of the oxygen tank, pressure gage, and tubing for charging a bomb calorimeter is illustrated in Fig. 226. A pellet press for compressing samples of standard calorimeter calibrating materials (page 222) into a suitable size to burn in the calorimeter crucible is shown in Fig. 227.

¹ ATWATER, *Bull.* 21, U. S. Dept. Agr.

Figure 225 shows another form of bomb calorimeter (Emerson) of which the Mahler is typical. It consists of a nearly spherical shell *S*, divided into two parts which are screwed together by the ring *R*. Powdered fuel is placed in the crucible *C* and is ignited electrically by the current passing through the water in the vessel *Q* from the terminal at *A*, then through an insulated contact point *P* in the bottom of the calorimeter to a small platinum or iron wire in the crucible *C*, which becomes heated by the passage of the current to a white heat, igniting the fuel. One end of this small wire is fastened to make electrical contact with the lining of the calorimeter, which in turn is connected electrically with the plug and terminal at *B*.

The outer vessel *O* is to be filled with water to the top. The stirring device consists of small propellers *F* on a vertical shaft operated by a small electric motor *M*.

The Emerson bomb is made of steel, with usually a lining of nickel, but gold or platinum lining is also used where heavy service is expected, because a nickel lining must be frequently renewed. There are two forms: the single valve, which is the one generally used, and the double valve, which has the advantage of being suitable for the removal of the products of combustion so that they may be analyzed. When samples of solid fuels having low volatility are to be tested, a small amount of naphthalene is put into the platinum fuel pan to act as a starter; and when this is done, a fuse wire of larger diameter is used than for ordinary service. For this bomb calorimeter, three types of jackets are obtainable. Figure 225 shows the regular jacket space filled with water. Another type of jacket is the so-called "adiabatic" kind (page 224), in which the jacket water is heated electrically at such a rate that the jacket-water temperature is approximately the same as that of the calorimeter water. A third type has an enclosed jacket in which there is a vacuum, the construction being similar to that of an ordinary thermos bottle. It is plain that when the "adiabatic" or the vacuum jacket is used the radiation effect is so small that, if the temperature of the calorimeter water is close to that of the room and the jacket at the beginning of a test, no radiation correction is needed.

Method of Testing with Bomb Calorimeter.—In the testing of solid fuels, it is customary to use a 1-gram charge of the air-dried¹ fuel sample,

¹ The Power Test Committee of A.S.M.E. recommends that the sample should be "air dried" at 240 to 380°F. (see p. 263), and that a similar "air-dried" sample should be tested for moisture, so that the final result may be based on heat value per pound **dry** coal. Many chemists and engineers prefer to use a sample of coal powdered "as received" and determine the heat value per pound "as received." The latter method does not give as accurate results as the former but slightly reduces the time required for making the calorific determination.

that is ground to such a size that it will pass through a 60-mesh¹ sieve. The sample of fuel to be tested is usually weighed in a flat-bottom platinum tray, with an acid-washed and previously "ignited" disk of asbestos paper interposed between the sample and the tray, the asbestos paper being used to avoid too rapid conduction of the heat of combustion and consequent incomplete combustion of the fuel. A fuse wire (preferably 0.1 millimeter in diameter) of platinum² wire is attached to the bomb electrodes so loosely that the middle part of the wire will dip down into the fuel in the platinum tray which rests on a platinum or a suitable alloy support provided for the purpose.

The advantage of using the platinum wire is that it does not burn at the temperature developed in the bomb, where an iron wire is always completely burned and a correction, therefore, must be made for the heat that results from the burning of the iron wire. The wires used for producing the combustion by electric heating called fuse wires should be of uniform length (about 2½ inches); and if iron wire is used, its length must be accurately measured and recorded, and its heat of combustion calculated. When the bomb is opened after combustion of the fuel sample, the fuse-wire residue (if iron) is weighed so that the amount used for kindling can be estimated. For every milligram of iron wire burned, 1.6 calories are subtracted from the heating value of the fuel.

The bomb should be carefully washed with distilled water, and the collected washings are to be *titrated* with standard ammonia solution (methyl orange or methyl red indicator being used) each cubic centimeter³ of which contains 0.00587 gram of anhydrous ammonia, NH_3 . For every cubic centimeter of this standard solution used, 5 calories are deducted from the determined heating value. In addition to the foregoing corrections it is necessary to apply another subtractive correction amounting to 13 calories for every 1 per cent of sulphur that is contained in the fuel.⁴

After the platinum tray has been placed inside the bomb with the fuse wire dipping down into the sample of fuel, the cover of the bomb is screwed tightly to the bottom part, and the valve in the cup intended for

¹ A 60-mesh cloth (usually made of wire) is the trade name for a fabric that has 60 meshes to the inch or, in other words, has 60 wires all in one direction in a 1-inch length. In other words, it has 60×60 or 3,600 meshes to the square inch.

² Instead of the platinum wire, a pure iron wire No. 34 B. & S. gage may be used.

The calorific value of pure iron is about 3,000 B.t.u. per pound, and iron wire No. 34 B. & S. gage 1 inch long will generate in combustion 0.63 B.t.u. This is the size of wire generally used in calorimeter work.

³ For explanation of this correction see *Bur. Mines, Tech. Paper 8*, p. 30.

⁴ The heat liberated in the formation of the acids is usually less than ¼ per cent, which, of course, would be subtracted from the calorific value obtained. If the reader is interested, he will find the method explained with the necessary data in the H. Poole, "Calorific Power of Fuels," p. 62.

the admission of oxygen gas is opened so that there is a flow of commercially pure 99 per cent oxygen¹ into the interior of the bomb.

The admission of oxygen gas to the bomb should be carefully watched to be sure that the gas does not enter too rapidly. Too quick entrance of the gas into the calorimeter is likely to blow some of the fuel dust out of the platinum tray and thus prevent the combustion of all of the sample. The next step is to put into the calorimeter bucket distilled water of the same weight that is used in determining the heat or water equivalent of the calorimeter. The temperature of the water² is to be adjusted so that, after combustion has taken place, the temperature of the water will not be more than 0.5°C. above that of the jacket water. The proper temperature difference to allow is best determined by experiment and for most of the commercial types of fuel calorimeters should be about 3°C., varying somewhat with the size of the calorimeter. When water of the required temperature has been placed in the calorimeter bucket, the bomb is placed in the water after the necessary electrical connections have been made. The next step in the arrangements for the test is to put in place the stirring device and the jacket cover, after which the calorimeter thermometer is set up in its proper position. Where the fuel calorimeter does not have an adiabatically controlled auxiliary water jacket or a vacuum-bottle jacket, precautions must be taken to be sure that the outside surface of the water container and also the walls of the "air" jacket are dry. Since these walls are approximately 3°C. warmer than the calorimeter, any moisture condensed there will evaporate and give up its latent heat of vaporization. A small amount of water evaporated in this way may introduce quite a large error in the results.

Specimen Test of Heat of Combustion of Solid Fuel.—The data in Table XV are taken from a test of solid fuel given in the A.S.M.E. Power Test Code on Heat of Combustion (Series, 1932, Part IX) to illustrate the method of calculating the heat of combustion (**higher value**, pages 243 and 311) from bomb-calorimeter readings.

Parr Calorimeter.—It is not always convenient to secure a supply of oxygen under pressure for use in a Mahler bomb, and consequently another type of fuel calorimeter, known as Parr's, using a chemical (sodium peroxide) as the source of oxygen, has found considerable use, especially for relative determinations in power plants. The results obtained can scarcely be depended on to be as accurate as determinations

¹ The oxygen gas must obviously be free from any *combustible* gas which, in burning, would add its heat to that generated by the combustion of the sample of fuel.

² It is difficult to obtain water at the exact temperature required, so that a convenient method in this case is to pour into the calorimeter bucket the required amount of water at a slightly higher temperature than that desired and to make the adjustment for the exact temperature by holding a small piece of ice in the water and stirring it with a thermometer used to observe the temperature.

TABLE XV.—METHOD OF CALCULATING FROM CALORIMETER READINGS¹

Sample No. Weight of coal, 1.0012 gram. Weight of water in calorimeter vessel and water equivalent of calorimeter, 3,000 grams.

Time,	Readings, Degrees	
P.M.	Centigrade	
1:54	23.874	} + 0.0058* rate of change in preliminary period
1:55	23.879	
1:56	23.885	
1:57	23.892	
1:58	23.897	} $\begin{cases} +a \ 0.0058 \\ +b \ 0.0027\dagger \end{cases}$
1:58½	24.160	
1:59	25.430	} $\begin{cases} +a \ 0.0008 \\ -b \ 0.0006 \end{cases}$
1:60	26.280	
2:01	26.439	} $\begin{cases} -a \ 0.0025 \\ -b \ 0.0026 \end{cases}$
2:02	26.463	
2:03	26.466	} $\begin{cases} -a \ 0.0026 \\ -b \ 0.0026 \end{cases}$

0.0066 algebraic sum, in degrees Centigrade, of radiation loss.

a Computed rate per minute of temperature change of each reading.

b Temperature correction for heat loss during each interval.

¹ This is the method recommended by the A.S.M.E. and is based on that described in Bur. Mines *Tech. Paper* 8. The U. S. Bureau of Standards method may be used if preferred. This is given in the American Society for Testing Materials book of standards for the year 1930, p. 724.

* The reading 23.874°C. taken at 1:54 p.m. is the first reading of the preliminary period. The temperature reading 23.897°C. is the last reading of the preliminary period and was taken at 1:58 p.m., 4 minutes after the first reading; hence 23.897 - 23.874 or 0.023°C. is the total change in temperature during the preliminary period; also 0.023 divided by 4 or 0.0058°C. is the rate of change per minute during the preliminary period.

† The temperature correction for the heat loss by radiation during the half-minute readings is made by dividing the algebraic sum of the rates by 4, and for the minute readings the algebraic sum is divided by 2, because the temperature correction for heat loss during each interval is the mean rate per minute of temperature change multiplied by the time of the interval in minutes. For example, at 1:58 p.m., the rate per minute of temperature change is +0.0058°C., and at 1:58½ the rate per minute of temperature change is +0.0049°C. Therefore, the mean rate per minute of temperature change for the half-minute interval is [0.0058 + (+0.0049)] ÷ 2 = 0.0054°C.; and ½ × (+0.0054) = +0.0027°C. is the correction for radiation heat loss taking place during the half-minute interval.

‡ The change in rate per degree of temperature change in the combustion period is 0.0084 ÷ 2.564 or 0.0033°C. The change in temperature during the first half minute of the combustion period is 24.160 - 23.897 or 0.263°C., and 0.263 × 0.0033 = 0.0009°C., which is the change in rate during the first half minute of the combustion period; also 0.0058 - 0.0009 = 0.0049°C., which is the rate of temperature change for the reading at 1:58½ p.m. The rate of change for each succeeding reading during the combustion period is calculated in the same way.

TABLE XV.—METHOD OF CALCULATING FROM CALORIMETER READINGS (*Continued*)

2:04	26.463	26.463 - 26.450 ÷ 5 or 0.013 ÷ 5 = 0.0026§ is rate of change in final period, degrees Centigrade
2:05	26.460	
2:06	26.458	
2:07	26.455	
2:08	26.454	
2:09	26.450	

	Temperature, Degrees Centigrade
Beginning.....	26.463
End.....	23.897
Observed temperature change.....	2.566
Thermometer correction.....	0.002
Stem correction.....	0.000
	2.564
Radiation loss.....	0.0066
	2.5706
Water and heat equivalent of calorimeter, grams (see p. 222)	3000
Total heat developed, calories.....	7,711.8¶
Correction (see calculation below).....	40.3

Heat developed by combustion of sample, calories.....	7,671.5
Heat developed by combustion of 1.0000 gram of sample, calories (higher value, p. 311).....	7,662.3
Heat developed by combustion of 1 pound of sample, B.t.u.	13,793.0
	Calories
Wire burned = 11.2 mg.	= 17.9
Titration solution (1 c.c. = 5 cal.), 2.5 c.c. (p. 231) =	12.5
Sulphur (0.01 gram = 13 cal.), 0.0074 gram	= 9.9

	40.3
Room temperature.....	24°C.

§ The rate of change per minute in the final period is -0.0026°C.

|| The change in rate during the combustion period is 0.0058 - (-0.0026) or 0.0084°C. The observed change in temperature during the combustion period is 2.564°C.

¶ The algebraic sum of all the temperature corrections for heat loss during each interval is added to the observed temperature change after corrections have been made for the errors and stem correction of the thermometer. This number when multiplied by the heat equivalent or water equivalent of the whole apparatus including the water used gives the total heat developed in the combustion.

with one of the bomb type.¹ Figure 228 shows one of the latest designs of the Parr calorimeter.

For accurate work with a Parr bomb calorimeter, it is essential that the sodium peroxide should be kept free from moisture, as this chemical

¹ The accuracy of the determinations depends very much on the chemical purity and dryness of the sodium peroxide. If it is absolutely pure and the apparatus is handled with skill, it is possible to get results comparable with the bomb types.

absorbs water readily. The effect of water absorption by sodium peroxide is to give too high a temperature rise in the calorimeter.¹

The bomb is fitted with removable propeller blades *W* which act as a convenient stirring mechanism. The cover which has a long stem *L* contains one of the electrodes *J* for electric ignition. This cover is screwed down on the bomb by the nut *F* which bears on an intervening gasket *G*. The long stem of the bomb cover is set up sufficiently high so that a driving pulley *P* can be attached to the top outside the calorimeter, and it may be revolved by a suitable electric motor which will revolve the bomb with its attached stirring blades on the pivot *F* (Fig. 229) at about 150 revolutions per minute.

Formerly the Parr bomb calorimeter was made

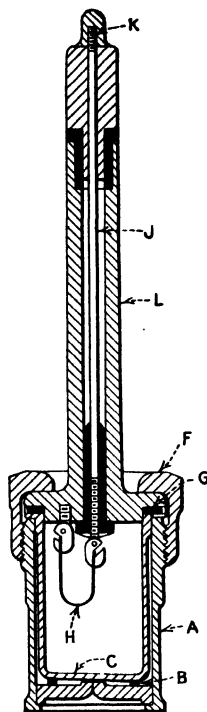


Fig. 228.—Parr calorimeter for electric ignition.

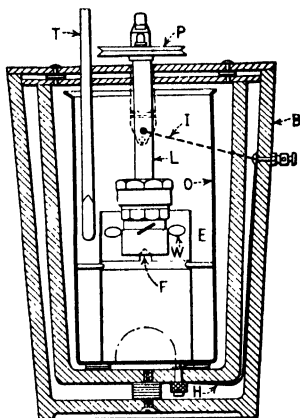


Fig. 229.—Parr fuel calorimeter showing stirring device.

with a hollow stem on the cover in which there was a valve at the bottom (Fig. 230). By this arrangement, a piece of heated iron could be dropped through the valve *M* for the ignition of the mixture of solid fuel and sodium peroxide.

An important precaution in the use of this calorimeter is that the moisture of this solid fuel and sodium peroxide should be thoroughly mixed. To accomplish this, the Parr bomb, as now made, is furnished

¹ Sodium peroxide must always be handled carefully, as it may react at ordinary temperatures in the presence of moisture to injure any kind of organic matter and may also develop enough heat to produce combustion and sometimes explosions. Because of this danger of explosion, unburned mixtures of fuel and sodium peroxide that must be disposed of should be slowly and carefully poured into water and should not be thrown into waste receptacles or ash cans. This chemical causes painful burns if it comes into contact with the skin.

with a false cover. The mixing is done by screwing in place the false cover and then shaking the bomb very thoroughly. After the shaking process, the bomb should be given a sharp tap on a hard surface at the bottom to disengage any part of the combustible mixture that might adhere to the false cover. After this mixing, the nut of the bomb is

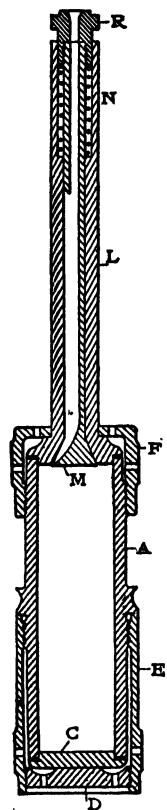


FIG. 230.—
Parr bomb for
hot tube igni-
tion.

unscrewed so that the false cover may be removed and the proper cover with its long stem *L* put in its place. A charge for the bomb consists of about 0.002 to 0.004 pound of pulverized coal from which the moisture has been driven off by warming for about an hour at a temperature of about 240 to 280°F. and **eighteen** times as much by weight of sodium peroxide, which supplies the oxygen needed for combustion. Reactions produced by combustion are complex, as the products of combustion CO_2 , H_2O , SO_2 , SO_3 , etc., combine with some of the Na_2O_2 or Na_2O to form Na_2CO_3 , NaOH , Na_2SO_4 , etc. The cover must be attached very securely by turning up the nut with a long wrench while the bomb is held in a vise or in some similar manner, because there is a violent explosion when ignition takes place.

When the hot-wire method (Fig. 230) is used, the wire must be put into the tube in the long neck *L*, the cap *R* at the top must be struck quickly with a mallet in order to open the valve *M*, which opens inward into the bomb and permits the wire to fall through before it cools. To be certain of obtaining a good result, the wire should be heated almost to a white heat, and the coal should preferably be put into the bomb after the sodium peroxide. The rise of the mercury in the thermometer will indicate when an explosion has occurred.

The water **equivalent** of the calorimeter is determined in the same way as for other fuel calorimeters. The recent designs of the Parr bomb calorimeter are so standardized in detail that each one has a constant water equivalent value of 3,100 grams. For a most successful operation, it is recommended that the water in the calorimeter at the beginning of a test be about 0.02°C. below that of the room. The preliminary temperature observations to determine the initial radiation rate as for other bomb calorimeters is explained on page 233.

The temperature rise during the period when the radiation rate is being determined should not exceed 0.01 to 0.02°C. per minute. It may usually be assumed that, if the temperature rise during this period is considerably more rapid or is irregular, there is likely to be leakage in the bomb. In case such leakage is discovered, the bomb should be imme-

diately removed from the water in the calorimeter and emptied as rapidly as possible in order to avoid an explosion. Under satisfactory conditions, the temperature of the calorimeter water rises rapidly during combustion and reaches its maximum reading usually after about 2 minutes, after which the readings should be continued for 5 or 6 minutes to obtain the final radiation rate.

The bomb is next removed, dried, and opened. The inner cup *C* with its contents is then placed in a large vessel containing about 500 cubic centimeters of water in which the cup should remain until all the fused peroxide is dissolved. It should then be removed, cleaned, and dried. The alkaline solution in the larger vessel should be tested for unburned carbon. This alkaline solution is acidified so that the black flakes of iron and copper oxide disappear, and the solution turns yellow with only the unburned carbon in suspension. The amount of this should be determined by filtration, drying, and weighing. A subtraction of 8.1 calories must be made for each milligram of unburned carbon. *The usual procedure, however, is to examine the bomb contents by eye and to omit the carbon-residue determination if none appears to be present upon such examination.*

Allowance must be made in the calculations for the heat generated by the chemical reactions of the sodium peroxide, which for the proportions given for a charge is approximately 27 per cent of the heat generated.

Example Illustrating Calorific Determination with Parr Calorimeter.—

In the following example the calorific value of a sample of powdered coal weighing 0.0040 pound, containing 1.3 per cent moisture and 6.7 per cent ash, is calculated directly in British thermal units per pound of fuel.

Weight of sodium peroxide.....	0.072 pound
Weight of water.....	7.160 pounds
Water equivalent of bomb.....	0.253 pound
Total equivalent weight of water.....	7.413 pounds
Temperature rise.....	10.2°F.
Total heat generated, 7.413×10.2	= 75.61 B.t.u.
Heat due to combustion of iron wire (see p. 231).....	1.11 B.t.u.
Heat due to coal alone $(75.61 - 1.11)(1.00 - 0.27)$	= 54.39 B.t.u.
Heat value per pound coal as fired.....	$\frac{54.39}{0.0040} = 13,598 \text{ B.t.u.}$
Heat value per pound dry coal.....	$\frac{13598}{0.987} = 13,780 \text{ B.t.u.}$
Heat value per pound combustible.....	$\frac{13780}{0.933} = 14,780 \text{ B.t.u.}$

Parr gives the following additional corrections to be applied to a given test in order to insure highest accuracy. Since the present commercial form of this calorimeter has a *standard water-equivalent value*, these corrections may be expressed in terms of temperature. The total corrected

temperature rise is first to be multiplied by 0.73; next apply the following deductions:

Percentage ash is multiplied by.....	0.0025
Percentage sulphur is multiplied by.....	0.0050
For 1 gram (0.002205 lb.) KClO_3 deduction equals.....	0.1500
Hydrogen factors:	
For all bituminous coals.....	0.0400
For black lignites.....	0.0560

Determination of Heating Value of Liquid Fuels with Bomb Calorimeter.—Liquid fuels of low volatility, such as fuel oils, may be tested for heating value in a bomb calorimeter by being burned in a tray or crucible in the same way that coal is handled. For the reason that the heating value of most liquid fuels is high, it is usually best to use, especially in the case of oils, samples that weigh about 0.8 gram, instead of 1.0 gram, as in the case of solid fuels (page 230). The weighing of the samples should, of course, be accurate, with determinations to the nearest 0.1 milligram. Quartz-glass crucibles¹ are recommended by the Power Test Committees of the A.S.M.E., to be used instead of platinum trays for the determination of the heating values of samples of oil.

If the liquid fuel that is to be tested for heating value has high volatility, such as gasoline or alcohol, it is very difficult to get satisfactory determinations if these liquid fuels are burned in a tray or crucible in a bomb calorimeter, the difficulty being that accurate weighing of these highly volatile liquids is almost impossible. The best method in this case is to use a small glass bulb having a capacity of about 2 cubic centimeters and a capillary stem about 3 centimeters long. This glass bulb should have very thin walls. After it is weighed, it should be gradually heated to expel a large part of the contained air. While the bulb is still warm, its capillary tip should be dipped down into the sample of liquid to be tested for heating value, and then the bulb should be allowed to cool in this position. After cooling, a sufficiently large sample of the liquid fuel will be drawn into the bulb, which may then cool in this inverted position. The capillary tip of the glass bulb must be closed by touching it to a hot flame. The bulb and the enclosed sample of liquid fuel must then be again weighed. The difference between this weight and that of the empty bulb gives the weight of the sample. The combustion tray is next placed in its proper place in the bomb calorimeter with the usual length of platinum or other fuse wire dipping into about 0.02 gram of standard naphthalene (page 224). The naphthalene is used as a quick-combustion means of breaking the glass bulb and igniting the sample of oil. The heat

¹ A wide-form crucible of quartz glass having a capacity of about 5 cubic centimeters may be conveniently used.

of combustion as determined in this way must be corrected by subtracting from it the heat that is generated by the burning of the naphthalene.¹ This procedure is generally considered rather difficult, and consistently accurate results are not easy to obtain.

Calorimetric Determination of Heating Value of Gases.—The heating value of gases is usually obtained with a type of calorimeter which provides a constant-pressure flow both of the water for the cooling medium and of the gas to be tested. The essential equipment for this kind of testing consists of the constant-flow calorimeter, a laboratory type of test gas meter, a gas-pressure regulator to provide gas at constant pressure, accurate calorimeter thermometers, a suitable burner that will give perfect combustion of the gas, and a simple hygrometer (with wet- and dry-bulb thermom-

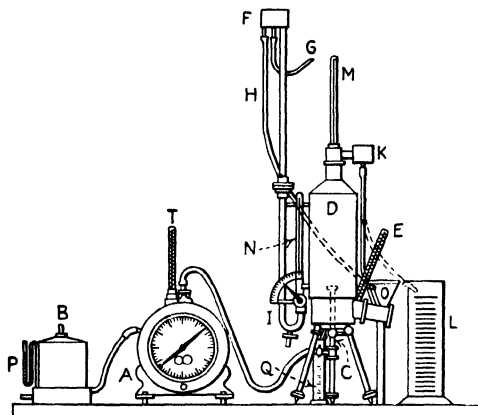


FIG. 231.—Junkers calorimeter with auxiliary apparatus.

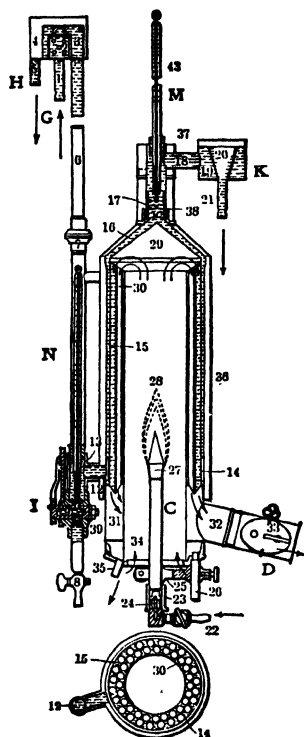


FIG. 231a.—Section of Junkers calorimeter.

eters). For determining the heating value of a gas, the Test Code of the A.S.M.E. recommends the use of "one of the many modified forms of the Junkers flow calorimeter." The original Junkers calorimeter, shown in Figs. 231 and 231a, received first mention in the list of approved apparatus for this kind of testing.

Junkers Calorimeter for Gases.—Apparatus for determining the calorific value of gases are shown in Figs. 231a and 232. The gas flowing in a pipe at the left (Fig. 231) passes through the meter A, then through the regulator B, and is burned in a type of Bunsen burner C in the lower

¹ The heat of combustion of naphthalene is explained on p. 222. If an iron fuse wire is used, correction must be made for its combustion as stated on p. 231.

part of the calorimeter. This instrument consists of a cylindrical copper vessel through which water is constantly circulating. The gases from the Bunsen flame in the calorimeter pass up through the hollow central portion of the instrument and near the top are deflected downward through a group

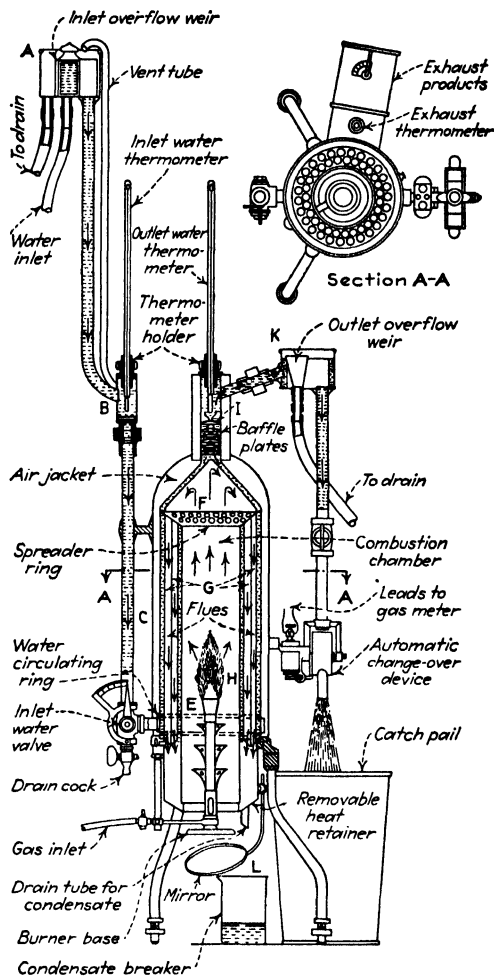


FIG. 232.—Sargent type constant-pressure calorimeter.

of small tubes arranged in an annular ring between the outside and inside walls of the calorimeter.¹ Around these tubes water is kept circulating

¹ A modification of Junkers calorimeter is made by the American Meter Company of New York and Philadelphia. In principle it is exactly the same as the one described but is much improved in mechanical construction, the greatest advantage being that the nest of small tubes can be readily removed for repairs. In the original design removal for repairs is difficult if not almost impossible. The "American" type is used exclusively by the U. S. Bureau of Standards, which is a sufficient guarantee of reliability.

continuously to absorb the heat generated by burning the gas tested. After leaving these tubes, the products of combustion discharge first into a chamber 31 (Fig. 231a) and then into the air through the flue *D*. In order to keep the flow of water as regular as possible, it is brought from the supply pipe *G* into a small reservoir in which the water is kept at a constant level (constant head) by means of an overflow pipe *H*. The water supplied to the calorimeter passes down through the pipe 6, through a valve at *I*, and discharges at *K*, running into a vessel in which it is weighed. A graduated tube *Q* (Fig. 231) is provided to collect the moisture from the steam that is condensed. The condensed steam collects in the combustion chamber 31 and escapes through the tube 35. A thermometer *N*, in a cup near the valve *I*, indicates the temperature of the water entering the calorimeter, and one at *M* shows the temperature of the water leaving. The temperature of the products of combustion (burned gases) is indicated by the thermometer *O* in the gas flue. The calorimeter is provided with an air jacket and is covered with sheets of copper, nickel-plated and highly polished, so that the radiation loss is considered negligible. If, then, the flow of water and the rate of burning the gas are regulated so that the **temperature of the products of combustion as indicated by the thermometer at *O* is the same as the temperature of the air surrounding the calorimeter**, practically all the heat generated by the burning gas is absorbed by the water. The rise in temperature of the water is observed by reading the thermometers at *N* and *M*.

Now if the temperatures of the water at the inlet and at the discharge have been observed and the weight of the water flowing has been determined while, for example, a cubic foot of gas has been burned, then the difference in temperature in degrees Fahrenheit times the weight of water in pounds gives the heat value in British thermal units per cubic foot of gas. This is called the **higher heat value** of the gas.

The Sargent modification of the Junkers constant-pressure flow calorimeter, shown in Fig. 232, has rapidly been coming into engineering service. It is one of the modified designs of the Junker type that is recommended by the A.S.M.E. The description going with Figs. 231 and 231a should make the operation of the Sargent apparatus sufficiently clear, so that no additional explanation is needed.

Results of calorific determinations of a gas should be stated in a report as calculated in heat units per cubic foot of gas for the **standard conditions** of pressure and temperature. The A.S.M.E. has favored the adoption¹ of **29.92 inches of mercury pressure and 68°F.**, while chemists and European engineers use as standard 29.92 inches (760 millimeters) of mercury pressure and 32°F. (0°C.). This conversion can be

¹ *Jour. A.S.M.E.*, December, 1924, p. 923.

readily made because the volume of the gas is directly proportional to the temperature.

For some calculations relating to the efficiency of heat engines it is desirable to know the number of heat units representing the calorific value of the gas when the steam formed in the combustion is not condensed but is carried off with the products of combustion as is the case in practice. To determine this value, sometimes called the **lower heat value** of the gas, the heat of evaporation at atmospheric pressure¹ of the weight of condensed steam collected in the Junkers calorimeter must be subtracted from the value obtained by multiplying together the rise in temperature and the weight of water used. This correction is usually about 5 to 10 per cent, having, as a rule, the smaller value for producer gases with high percentages of carbon monoxide.

When thermal efficiencies of gas engines are calculated, it should always be clearly stated whether the "higher" or the "lower" heat value of the gas has been used. In all the codes of the A.S.M.F. the "higher" heat value has the preference, but this is not by any means the generally accepted practice.

This apparatus, although it operates by a constant-pressure method, gives very satisfactory determinations. Radiation loss is small and is neglected. Since tests with this apparatus are started when all parts are already heated normally, no water equivalent (page 222) is taken into account. If the temperature of the discharge gases is not the same as that of the air supplied, the results will be in error but the amount of this correction (see table on page 246) and the methods of computing even its approximate value are uncertain. For this reason the temperature of the discharge gases should be regulated carefully and not more than 2 or 3°F. difference should be permitted between the temperature of these gases and that of the air supplied. Often it is necessary to open the windows of laboratory rooms to secure the proper temperatures. Temperatures of the water should be practically constant before a test is started.

As the Junkers calorimeter is ordinarily operated, it does not determine accurately the **higher** heating value even if all the precautions stated have been observed, because an excess of moisture above that

¹ This correction can be computed with greater accuracy in terms of the following symbols: w = weight of moisture in gases, in pounds, h_g = the total heat of saturated steam at pressure and temperature of condensation, t_1 = temperature of condensed steam, in degrees Fahrenheit, t' = temperature at which condensation takes place, t_2 = temperature of the gases discharged from the calorimeter, C_s = average specific heat of steam from t_2° to t_1° (about 0.46), C_w = average specific heat of water from 32° to t_2° ; then, heat lost by gases, in B.t.u., in condensation of water vapor (h') = $w[H - C_w(t_1 - 32) + C_s(t_2 - t')]$; then if $C_w = 1$, and in a different form, $h' = (h_g - C_s t' + 32 - t_1 - C_s t_2)w$. Now the term $h_g - C_s t'$ has the approximately constant value of 1057 for a wide range of values of t' , so that $h' = (1057 - t_1 + 0.46t_2)w$.

which came in with the air goes off in the discharged gases. The only way to eliminate this error is to supply gas and air that are saturated with moisture. The calorimeter will then give the true "higher" heating value, because all the moisture resulting from the combustion of hydrogen will be condensed and will give up its latent heat. When a wet-gas meter is used, it may be assumed that the gas is saturated as it comes to the apparatus. The obvious way to eliminate this error is to supply air which is also saturated. A convenient design is to connect the closed top of a cylindrical vessel about 2 feet high and 5 inches in diameter by a 1-inch rubber tube to the bottom of the calorimeter, which, except for the opening for this tube, has been made air-tight. The cylinder is provided with a water-waste cock at the bottom. Several trays covered with coke are placed inside the cylinder, which is perforated with a number of half-inch holes around the perimeter near the bottom for the admission of air. A water jet discharges from the top of the cylinder and the water trickles down over the coke as the air enters at the bottom and passes up to the air pipe leading to the calorimeter. By this method air is thoroughly saturated and not only more accurate but also much more consistent results for the **higher heat values** are obtained. Obviously, this humidity correction has no effect at least in the theory of combustion on the **lower heat values**, as they are calculated for the condition when all the water vapor due to combustion leaves in the discharged gases. This is one reason why many engineers prefer to base calculations of thermal efficiency on the "lower" heat values.

The error due to humidity can, however, be calculated approximately and the results correspondingly corrected. Moisture carried in the air can be determined by a wet- and dry-bulb thermometer (see page 186) and then, if it is assumed the discharged gases and the gas burned are saturated, the excess of condensation carried away in the discharged gases is readily calculated since the weight of the gases can be determined by a laborious calculation involving the computation of the weight of air supplied, which must be obtained from the analysis of the products of combustion (discharged gases).

The "test" gas meter (Fig. 231) used for measuring the amount of gas used must, of course, be *calibrated* for important tests, preferably by means of a standard fractional cubic-foot bottle. A one-tenth or a one-twelfth cubic-foot bottle should be used, depending on the scale of the gas meter, that is, whether one revolution of the pointer over the dial is one-tenth or one-twelfth of a cubic foot. The fractional cubic-foot bottle consists of a metal globe provided with glass tubes, one at the top and another at bottom, each with a gage mark so placed that the volume included between the gage marks is exactly one-tenth or one-twelfth of a cubic foot, as the case may be. The glass tube at the top of the metal

globe has a three-way cock, one branch of which is connected to the gas supply and the other to the meter being tested. The lower glass tube is connected by a rubber tube to a reservoir containing water. Two shelves are provided by this reservoir, so that, when it rests on one, the water level will coincide with the gage mark on the bottom glass tube and, when it rests on the other, higher shelf, the water level will coincide with a mark on the glass tube at the top of the metal globe, the gas in the bottle and its connections being at all times at atmospheric pressure.

In the calibration of the gas meter with the standard fractional cubic-foot bottle, the following points should be noted: (1) The meter must be level, as will be shown by the leveling device provided. (2) The lower meniscus of the water level (page 4), as shown in the gage glass, must be at the marked level. (3) Determine next if the shelves provided for receiving the testing "reservoir" are of the right height so that the water level in the glass tubes coincides with the marks on these tubes (when pressure is atmospheric). (4) It is necessary that the water with which a "test" gas meter is partly filled and the water in the standard fractional cubic-foot bottle should be completely saturated with gas before the meter is used to measure the flow of gas for a calorimetric determination. (5) The water in the gas meter and in the standard fractional cubic-foot bottle must be at approximately the same temperature.

The following procedure is suggested for the calibration of the gas meter: (1) Connect one branch of the three-way cock to the gas meter, the other branch to the gas supply line, and attach a gas burner to the outlet pipe of the gas meter. (2) With the standard bottle containing some gas and connected to the meter, test for leakage by closing the meter outlet and raising the reservoir so as to place the gas confined in the standard bottle at a pressure of about 2 inches of water above atmospheric. The water level should be at the mark on the top glass tube; and if after 5 minutes the water level has not changed appreciably, the system is tight. (3) Bring the gas-meter pointer to zero by passing sufficient gas from the bottle and then connect the bottle to the gas supply. (4) Fill the bottle with gas to the lower gage mark, and disconnect the rubber gas-supply tube to bring the pressure within the bottle to atmospheric. This should not change the level at the lower gage mark. (5) Connect the bottle to the meter, raise the "reservoir" to the elevated position (top shelf), and allow the gas to discharge through the meter till the water level reaches the top gage mark. When this mark is reached, the meter reading should be accurately observed and recorded. Refill the bottle and displace the gas alternately, as described several times, noting for each case temperatures of room, water in meter, and in bottle.

If during the passage of gas through the meter the pressure at the inlet is appreciably above atmospheric, the volume passing through it

will be smaller than that measured by the bottle. Usually, the correction for such pressure is negligible.

Precautions in Use of Constant-pressure Flow Calorimeter.—The temperature of the water flowing through the calorimeter should be kept constant and should not vary more than 2°F . from that of the room. A good arrangement is to have available a large tank of water in the calorimeter room which can be kept standing in the room for some time before tests are to be made. The room temperature should not be subject to any considerable variations, usually not more than 2°F . Since the gas and air supplied to the calorimeter burner will be at room temperatures, variations in temperature lead to inaccuracy.

The combustion of the gas burned in the calorimeter must be complete. If the completeness of burning is doubtful, the products of combustion must be tested for the presence of carbon monoxide, CO , of which there should not be more than a trace in a satisfactory test.

In gas calorimetry, it is customary to take temperature observations in degrees *Fahrenheit* in distinction from calorimetry work with solid fuels, where the temperatures are in degrees *Centigrade*.

Sufficient time must be allowed for the calorimeter conditions to come to equilibrium before a test is started. With gas burning and water flowing at the normal test rate, this will require from 5 to 30 minutes, varying with the type of calorimeter. This time of waiting for constant conditions must be found out from actual tests.

Calorimeter thermometers should be calibrated and graduated for an accuracy of 0.1°F .

Before a calorimetric determination on a sample of gas is made, the gas meter should be checked as to (1) level of meter; (2) water level; (3) saturation of water, with the kind of gas to be tested; (4) absence of leaks of gas, freedom of gas remaining from previous test.

The water flow through the calorimeter should be started a little while before the test begins, so as to remove any air that might be trapped in the water circulating system of the calorimeter. The water supply of the calorimeter should be adjusted to give a temperature rise due to combustion of the gas of from 15 to 20°F .

Gas-pressure Regulator.—In the use of a gas calorimeter, it is necessary to use a suitable device for maintaining a constant pressure of the gas supplied for combustion. A suitable apparatus of this kind is shown in Fig. 233, which, in its essential parts, is similar to a gasometer (page 189). This gas regulator has a conical valve V at the end of a long vertical valve stem S , which is connected to the center of the dome D of the water-sealed drum. The position of the conical valve, with relation to its seat, is controlled by weights which are placed over the extension of the valve stem in such a way that they counterbalance the buoyancy of the drum. Any rise in pressure of the gas supplied to the regulator

TABLE XVI.—HEATING-VALUE TEST RECORD¹

Place:	Date:	Time:	Kind of gas:			
Gas meter No.....	Thermometer No.:	Inlet.....	Outlet.....			
Gas line purged: (Yes) Meter adjusted: (Yes) Leak test: (Yes) Water-valve setting:						
..... Calorimeter damper closed (Yes)						
Differential thermometer correction determined: (date) Last meter calibration: (date)						
	Start	End				
Temperature of barometer, °F.....	68	68				
Barometer reading, in. mercury.....	29.50	29.50				
Temperature correction of barometer (p. 4).....	— .10					
Corrected barometer height, in. mercury at 32°F.....	29.40					
Pressure of gas at meter, in. water.....	1.5					
Equivalent pressure, in. mercury.....	0.11					
Total gas pressure, in. mercury.....	29.51					
Meter therm. reading, °F.....	68.1	68.3				
Calibration correction, °F.....	—0.3					
Average corrected meter temperature, °F.....	67.9					
Temperature and pressure reduction factor <i>F</i> of meter (to standard conditions ²	0.986					
Psychrometer { dry-bulb temperature, °F.....	68.0	68.5				
{ wet-bulb temperature, °F.....	53.0	53.0				
Humidity, relative (pp. 186 and 442).....	35%					
Temperature of products of combustion.....	69					
Time of 1 meter revolution, sec.....	52					
Equivalent rate of meter, cu. ft. per hr.....	6.9					
	Test 1		Test 2	Test 3		
	Inlet	Outlet	Inlet	Outlet	Inlet	Outlet
Preliminary.....	67.94	86.54	67.93	86.40	67.90	86.40
		86.50		86.45		86.45
		86.50*		86.49*		86.48*
		86.43		86.40		86.40
		86.36		86.43		86.39
		86.43		86.47		86.43
		86.54		86.50		86.49
Used in averaging.....	67.93	86.49	67.93	86.50	67.90	86.52
		86.50		86.41		86.50
		86.50		86.40		86.40
		86.47		86.48		86.42
		86.40		86.38		86.45
Supplementary.....		86.49		86.41		86.46
	67.93	86.36	67.92	86.41	67.90	86.47

TABLE XVI.—HEATING-VALUE TEST RECORD (Continued)

	Test 1		Test 2		Test 3	
	Inlet	Outlet	Inlet	Outlet	Inlet	Outlet
Average temperatures, °F.....	67.93	86.46	67.93	86.45	67.90	86.45
Calibration correction of them., °F..	-0.18	-0.25		-0.25		-0.25
Differential thermometer correction, °F.....		-0.02	-0.18	-0.02	-0.18	-0.02
Emergent stem correction of thermometers, °F.....		+0.08		+0.08		+0.08
Corrected temperatures, °F.....	67.75	86.27	67.75	86.26	67.72	86.26
Temperature rise T , °F.....	18.52		18.51		18.54	
Water heated W , lb.....	6.74		6.74		6.72	
Number of revolutions of meter.....	2					
Meter calibration 1 revolution, cu. ft.	0.1006					
Gas volume V , cu. ft. (0.1006 $\times$ 2)...	0.2012					

Total Heating Value of Gas

Observed heating value			
$\frac{W \times T}{V \times F}$, B.t.u. per cu. ft.....	629	629	628
Correction for atmospheric humidity	+4	+4	+4
Total or higher heating value of gas, B.t.u. per cu. ft.....	633†	633†	632†
Average of heating value from three tests 633 B.t.u. per cu. ft. (std. conditions of 68°F., 29.92 in.)‡			

Net or Lower Heating Value of Gas

Condensed water collected:	Calibrations	
	No. 1	No. 2
Meter reading: start.....	18.1	19.4
end.....	19.1	20.4
Condensate cubic centimeters.....	21.6	21.2
Condensate (c.c.) per cu. ft. of gas at 68°F., and 29.92 in. (std. conditions).....	21.8	21.4
Average condensate per cu. ft. (4), c.c.....	21.6	
Observed heating value average.....	629	
Reduction to net ($A \times 2.265$)§.....	-49	
Net or lower heating value of gas B.t.u. per cu. ft.....	580†	

¹ Data sheet (somewhat abridged) suggested by the U. S. Bureau of Standards.

² Standard conditions of temperature (68°F.) and pressure (29.92 inches) are computed from the equation:

$$\text{Correction factor } F = \frac{(P - E) \times (68 + 459.6)}{(t + 459.6) \times (29.92 - 0.69)}$$

where P = total (absolute) gas pressure in inches of mercury.

E = vapor pressure of water at t °F., in inches of mercury (see table, p. 593).

t = temperature of gas, in degrees Fahrenheit.

* Readings of temperature taken every 10 seconds.

† Loss of heat to surroundings may usually be neglected. Its magnitude depends on the design of the calorimeter, the rate of gas consumption, and the rise in temperature of the water. It is usually from $\frac{1}{2}$ to $\frac{1}{4}$ per cent and is, of course, always additive.

‡ To convert to heating value of saturated gas at 60°F. and a barometric pressure of 30 inches of mercury (at 32°F.), multiply by the factor 1.0240.

§ Heat of evaporation, h_{fg} , of water at the temperature (67.9°F.) and partial pressure at which the gas was tested is 1.054 B.t.u. per pound (p. 593) or 2.265 B.t.u. per gram or per cubic centimeter (c.c. of water).

As stated by Dalton's law there is in every "mechanical" mixture of gases a pressure for each gas, which is called its *partial pressure*.

has the effect, therefore, of tending to raise the drum but at the same time to reduce the size of the conical opening for the flow of gas through the valve *V*. A reduction in pressure, on the other hand, has just the opposite effect, as it will tend to increase the opening for the flow of

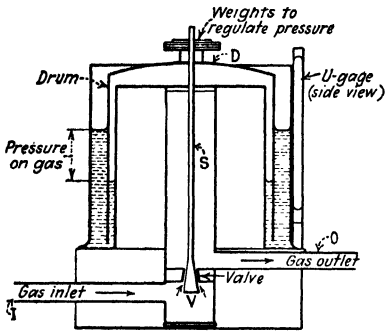


FIG. 233.—Weighted gas-pressure regulator.

gas through the valve. With the use of a device of this kind, fluctuations of gas inlet pressure are very much reduced and, if the fluctuations are small, they may be almost entirely eliminated.

The pressure at the gas outlet is a function of the weights placed on the drum and is measured by the U-tube shown in the figure by a side view. It is stated that devices of this kind are satisfactory for evening out the pressure of a gas supply even when that pressure varies several inches of water.

It is best to connect the gas regulator between the gas supply and the meter, in order to establish the certainty of constant conditions at the meter. If, however, the regulator is connected between the meter and the burner it is necessary to observe quite frequently the water level in the meter to make certain that it does not change, for, if the water level does change, the measurements by the gas meter would be in error by the amount of change.

For making determinations of the calorific value of suction producer gas where the working gas pressure is less than atmospheric, a good method is to collect a sample with an aspirator and a collecting bottle as explained for sampling flue gas (see page 268).

Producer gas and other gases of low heating value can be mixed with a little air and burned in a simple metal tube, covered over at the end with a piece of fine gauze to prevent firing back into the mixing chamber. The mixture formed should preferably be nonexplosive.

Recording Gas Calorimeters.—Many testing engineers prefer, for calorimetric determinations of the heating value of gas, a continuous recording instrument rather than the laboratory type. The best types of recording calorimeters do not, as a rule, give quite so accurate results as the laboratory kind. This statement is particularly true in cases where a highly skilled operator is not available to keep the recording instrument in proper adjustment. One of the disadvantages of the recording type of calorimeter is that it is very expensive. When the continuous instrument is changed from one kind of gas to another, an interval of about 30 minutes must be allowed before results that are indicated can be considered reliable (see Power Test Code of A.S.M.E., Part IX, page 16).

The author has made *continuous recording gas calorimeters* using a simple pipe burner and positive pressure blowers (similar in design to those on page 439). One blower for measuring the gas is of about one-sixth the capacity of the larger one for measuring the air. As both blowers are driven by a single electric motor, they will always deliver gas and air in a constant ratio, provided pressure and temperatures of gas and air are maintained at about the values at which the instrument was calibrated. Calorific value of the gas is then proportional to the difference in temperature between the air and gas entering and the temperature of the discharged gases. A differential recording thermometer with the chart graduated in heat units per cubic foot of gas as determined by comparison with a Junkers calorimeter gives a continuous record of heat values of the gas. An apparatus of this kind is very useful in a producer gas plant in showing the quality of gas produced and the relative care observed in the operation of the producers.

Exercise. Calorific Value of Gas.—One cubic foot of coal gas at an absolute pressure of 29.51 inches of mercury and at 67.9°F., when burned in a Junkers calorimeter, raised the temperature of 6.74 pounds of water in one test from 67.75 to 86.27°F. Condensation (water) collected due to combustion was 21.6 cubic centimeters; absolute atmospheric pressure was 29.40 inches of mercury; and corresponding heat of evaporation, h_{fg} , of water from tables is 2.265 B.t.u. per cubic centimeter (see footnote, page 247). The data and results of this test and two check tests are given in Table XVI (page 246).

Corrections for Difference between Inlet-water Temperature and Room Temperature.—The correction calculated from the following table¹ may be applied to heating values of illuminating gas of about 600 B.t.u. per cubic foot. The correction is added if the inlet water is warmer than the room, subtracted if the inlet water is colder.

TABLE XVII.A.—CORRECTIONS FOR DIFFERENCES BETWEEN ROOM AND WATER TEMPERATURES

Room temperature, degrees Fahrenheit	Corrections in B.t.u. per 1°F. difference	
	For calculating "higher" heating value	For calculating "lower" heating value
50	0.5	0.4
60	0.6	0.4
70	0.7	0.4
80	0.8	0.4
90	0.9	0.5
100	1.0	0.5

¹ From *Circ. 48*, U. S. Bur. Standards.

Junkers Calorimeter for Liquid Fuels.—Figure 234 shows a balance and lamp attachments for a Junkers calorimeter set up for determining the heat value of liquid fuels like gasoline, kerosene, crude oil. The heat generated is measured in the same way as when gas is burned and the

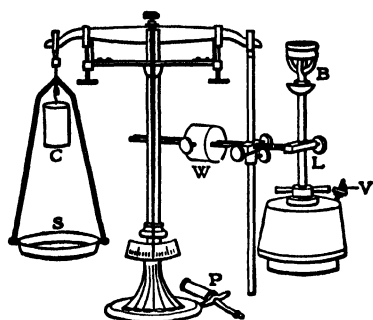


FIG. 234.—Balance and lamp for burning oils in Junkers calorimeter.

weight of oil used is determined by weighing on the balance to which the lamp *L* is attached. This lamp is provided with a “regenerative” burner *B* with a long stem as shown. A small hand pump *P* is arranged for attachment to the valve *V* to put the oil in the bowl of the lamp under a pressure of about 10 pounds per square inch. This air pressure forces the oil up the stem and through a coil of metal tubing which lies above the flame and is heated by it and gasified. The gaseous fuel escapes as a jet through a minute orifice where it should burn with a blue flame indicating perfect combustion. When oils heavier than gasoline are tested, the regenerator coil must first be heated by burning alcohol in the cup shown in the figure just below the burner *B*. When combustion is not complete, as is always the case when the flame is started, soot accumulates on the coil and is likely to choke the orifice of the burner. A piece of fine piano wire should always be at hand for cleaning the orifice.

The principal defect of this apparatus for the determination of the calorific value of liquid fuels is that there is sometimes leakage of the liquid fuel through the orifice of the burner into the cup located below it; and combustion in the cup is not likely to be complete. Another difficulty, which is probably even more important than the one already mentioned, is that the air capacity in the bowl of the lamp is so small that there is nearly always a considerable drop in the pressure on the liquid fuel in the bowl from the beginning to the end of a test run, the drop in pressure producing, of course, a corresponding reduction in the rate of flow of the liquid fuel through the burner. At Cornell University,¹ a constant-pressure device has been developed which can be very satisfactorily used

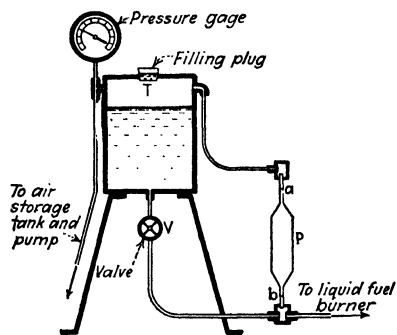


FIG. 235.—Volume measuring device for liquid fuel.

¹ DIEDERICHS and ANDRAE, “Experimental Mechanical Engineering,” p. 808.

to obtain a constant pressure on the liquid fuel. It consists of a metal tank *T* to serve as a reservoir of the liquid fuel, with a capacity of about 1 gallon, which should be fastened permanently to the table on which the calorimeter is to be used, somewhat as shown in Fig. 235. Connected to the bottom of the tank *T* is a small pipe to which is attached a valve *V*, and at one side of the tank is a pipette *P* for showing the level of the liquid fuel at the beginning and the end of a test run. Supplementing this 1-gallon tank on the table is a considerably larger tank on the floor below the table, in which a nearly constant pressure is maintained by means of a hand air pump, like those used for filling automobile tires. The air volume in the floor tank added to that in the tank *T* on the table is therefore rather large in comparison with that which can be obtained in the bowl of the lamp, and the pressure variation during a test can therefore be made almost negligible.

In the conduct of a test of calorific value of liquid fuel, as long as the calorimeter is being adjusted to obtain constant conditions, the valve *V* below the 1-gallon tank is open and the liquid fuel is supplied directly to the burner. On account of the large capacity, there is plenty of time to bring the calorimeter to equilibrium conditions. When these are reached, and it is desired to start the test, it is merely necessary to close the valve *V*. The time required to close this valve is immaterial, since the liquid fuel is now supplied through the pipette *P*, in which the level should be as yet above the upper mark *a*. When the level passes through the mark *a*, the test is started. It is ended when the level passes the mark *b*. The capacity of the pipette *P* is usually made about 20 cubic centimeters. The manipulation is thus much simpler than with the standard appliance. The only objection is that the measurement is in terms of volume which has to be converted to weight by a separate determination of the density of the liquid fuel.

Calorific Values from Chemical Analysis.—Dulong stated a long time ago that the heat generated by burning any fuel was equal to the sum of the "possible heats" generated by its component elements, less that portion of the hydrogen which combined with oxygen in the fuel to form water. When hydrogen and oxygen exist together in a compound in the proper proportions to form water, the combination of these elements has no effect on the calorific value of the compound. Now the calorific value of a pound of carbon is 14,600 B.t.u. and of a pound of hydrogen is 62,000 B.t.u., so that by Dulong's formula the calorific value of a pound of fuel *x* would be stated, using these values, as

$$x = 14,600C + 62,000\left(H - \frac{O}{8}\right) + 4,000S, \quad (59)$$

where *C*, *H*, *O*, and *S* are, respectively, the weights of the carbon, hydrogen, oxygen, and sulphur in a pound of fuel.

A similar formula known as that of the *Verein deutscher Ingenieure* expressed in units and terms used in Dulong's but corrected for the weight in pounds of the moisture w in a pound of fuel, is given as follows:

$$x = 14,400C + 62,000\left(H - \frac{O}{8}\right) + 4,500S - 1,100w. \quad (60)$$

Formulas given above are all for the "higher" heat values corresponding to those obtained with a bomb calorimeter. The last formula [Eq. (60)] expressed for "lower" heat value is

$$x = 14,400C + 52,000\left(H - \frac{O}{8}\right) - 1,100w. \quad (61)$$

As the result of testing 44 different kinds of coal with his bomb calorimeter, Mahler developed the following formula, using the same symbols as in Dulong's, in percentages instead of fractions of a pound,

$$x = 200.5C + 675H - 5,400 \quad (62)$$

Using this latter formula, Lord and Haas¹ computed the calorific values for a series of 40 Pennsylvania and Ohio coals which they had analyzed and found that the maximum differences between the calculated results and the determinations with a bomb calorimeter were from 2.0 to -1.8 per cent. With fuels like coke, charcoal, and anthracite coal, in which the content of volatile matter is small, the calorific values calculated from an accurate analysis are usually in very close agreement with accurate calorimeter tests, but with coals having more than 20 per cent of volatile matter there is likely to be considerable error.

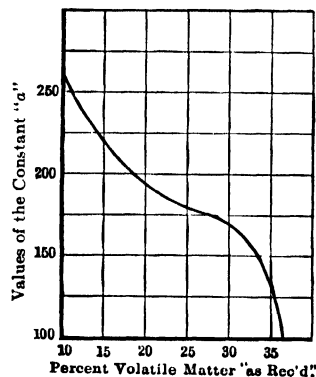


FIG. 236.—Curve for determining calorific value of coal.

A formula which is as accurate as any of those given above is based on the results of the proximate analysis. In this formula, known as Goutal's,²

c = percentage fixed carbon in coal "as received."

v = percentage volatile matter in coal "as received."

a = constant from the curve in Fig. 236.

Then,

$$x = 147.6c + a \times v. \quad (63)$$

Determination of Total Carbon.—In practical work it is difficult and expensive to determine the ultimate analysis of a sample of coal. If, however, in addition to the heating value, the total carbon content is known, a

¹ *Trans. A.I.M.E.*, February, 1897.

² *Wisconsin Eng.*, December, 1911.

complete and satisfactory heat balance of the combustion can be computed.

The carbon content of coal may be determined by using the apparatus shown in Fig. 237,¹ used in conjunction with a Parr fuel calorimeter. In the sodium peroxide fusion in the calorimeter, the total carbon of the coal has been oxidized and chemically combined to form sodium carbonate, NaCO_3 . By liberating the carbon dioxide, CO_2 , under accurately determined conditions of temperature and pressure in this apparatus, the amount of carbon may be derived from the volume of carbon dioxide, CO_2 , discharged. The large double pipette *P* is one-half filled with a 40 per cent solution of sodium hydrate, or such as is ordinarily used for the absorption of carbon dioxide gases. The sodium peroxide is dissolved in hot water and placed in the flask *F*. The solution in the flask is boiled to dispel the air. Concentrated hydrochloric acid is added through the funnel-shaped vessel *V*. The acid unites with the sodium carbonate, NaCO_3 , forming sodium chloride, NaCl , carbon dioxide, CO_2 , and water, H_2O . The carbon dioxide passes through the condenser *C* into the burette *B*. The water and the acid are condensed by the condenser *C* and remain in the flask *F*. The gas which collects in the burette *B* is then passed into the double pipette *P* containing the caustic solution which absorbs the carbon dioxide. The gas is then drawn back into the burette and the difference in volume is the volume of carbon dioxide absorbed by the caustic solution. From the carbon dioxide liberated in this manner, the weight of carbon in the coal may be readily computed.²

Proximate Analysis of Coal.—For all tests in which an analysis of the coal or its calorific value is to be determined, it is very necessary that the sample to be tested be selected with the greatest care. The method generally adopted for obtaining a fair sample is known as "quartering." This method as practiced by the U. S. Bureau of Mines is shown in Fig. 238 (page 260). The utmost care must be taken that the amount of moisture in the sample received for analysis is the same as that in the original condition, or, more specifically in a **boiler test**, at the time when the coal used in the test was weighed. For this reason samples of coal should

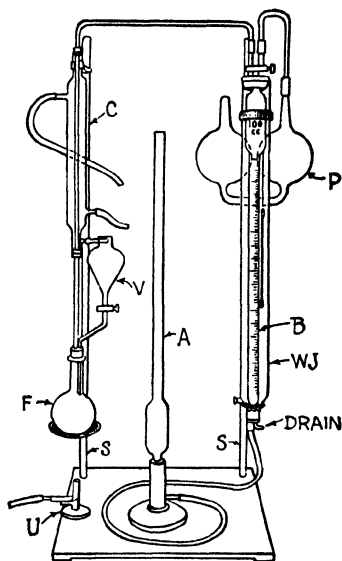


FIG. 237.—Apparatus for determination of total carbon.

¹ See Parr, "Fuel, Gas, and Lubricants."

² See Allen and Bursley, "Heat Engines," 4th ed., p. 117.

be transported and stored in air-tight preserving jars or similar vessels. It is not unusual, moreover, to find that coal containing 10 per cent of moisture will lose as much as 2 or 3 per cent of its moisture in the process of careless sampling, crushing, resampling, etc.; while if it is allowed to remain exposed to atmospheric conditions for a considerable time in a warm room, as much more may be lost by evaporation. Crushing, sampling, and weighing should always be done as rapidly as possible.

Only during recent years has the proper importance of correct sampling of coal for analysis been understood. Particularly in run-of-mine coal, that is, coal as mined without crushing or screening, the careful selection of coal for the sample as regards size is also very important. If the sampling has not been done so as to get coal that is representative, certainly the analysis can be of no value. Proportionate amounts should be taken of both large and small sizes as well as of the fine dust.

Details of solid fuel sampling as required by the A.S.M.E. Test Codes are explained on page 258.

In the laboratory the coal should be emptied from the jar or can and crushed to a fineness of about a 20-mesh sieve (20 meshes to the inch). The crushed coal is thoroughly mixed and a small portion, about 2 or 3 ounces, is put into an air-tight bottle and is to be used for the analysis. The rest is put back into the jar or can and sealed. It is to be retained for possible use in check tests.

Proximate analysis of coal consists in determining the moisture, volatile matter, fixed carbon, ash, and sulphur. Methods for these determinations are more or less empirical and vary slightly, so that in a report the authorship of the methods used should be stated. The methods most generally accepted by progressive engineers are those defined by the American Chemical Society¹ and are in common use both in this country and in England.²

Moisture and ash determinations are most important because they are noncombustible and detrimental constituents, the moisture requiring the wasting of heat for its evaporation into steam and the ash when present in large amounts is often likely to form clinker in poorly designed or badly operated furnaces and is also expensive to dispose of. Sulphur determinations are important only when the furnaces are not suitable for the combustion of coal containing 1 or 2 per cent of sulphur. Furnaces are readily designed for burning coals having from 4 to 5 per cent of sulphur without clinkering the ash.

Methods of the American Chemical Society are as follows:

1. **Moisture.**—Weigh in a covered crucible about 4 grams (about $\frac{1}{8}$ ounce) of the coal passing through a 20-mesh sieve that was prepared for analysis as described above. This should be done as quickly as

¹ *Jour. Am. Chem. Soc.*, Vol. 21, 1899.

² Royds, R., "The Testing of Motive Power Engines," p. 293, London, 1911.

possible to avoid loss of moisture to the air. Remove the cover and heat in an oven for an hour at a temperature of from 220 to 230°F. At the end of the hour replace the cover on the crucible, remove it from the oven, and place it in a desiccator to cool.¹ When the crucible and the coal it contains are at nearly the same temperature, they should be weighed. Again remove the cover and heat as before in the oven for a half hour longer. If the weight has remained constant, no more heating is necessary and the difference between the first and last weighings is the moisture in the coal.

2. Volatile Matter.—Weigh about 1 gram ($\frac{1}{80}$ ounce) of the “20-mesh” crushed coal prepared for analysis in a platinum crucible weighing from 20 to 30 grams ($\frac{2}{3}$ to 1 ounce)² and having a closely fitting cover. Support the covered crucible on a chemist’s triangle of nichrome steel or platinum about 3 to 3½ inches above the top of a good Bunsen burner.

Heat the covered crucible for **7 minutes** in the full flame of the burner, which should be at least 8 inches high when unobstructed. Cool the crucible in a desiccator and then weigh carefully. The loss is the sum of the volatile matter plus the moisture. The room in which the test for volatile matter is made should be free from drafts which might cause a variation in the intensity of the flame.³

3. Fixed Carbon and Ash.—The crucible without the cover and the residue from the preceding test are now intensely heated with a Bunsen flame or with an air-blast lamp until all the carbon has been burned, and the weight of crucible and contents becomes constant. The use of the air-blast lamp in place of the Bunsen burner for this last determination will very much reduce the time required. The contents of the crucible may be stirred slightly with a platinum wire to break up the ash, which

¹ While the sample is cooling, there is the possibility that it may absorb moisture from the air unless it is placed in a desiccator until cool. It is difficult to get accurately the weight of hot bodies on account of the air currents produced.

² The capacity of the crucible should be about three times the volume of the coal, to allow for its expansion in coking.

³ It is not at all unusual for persons making tests on the same sample of coal to disagree as to the volatile content. This is because not all chemists and engineers will use the same type of burner, and if they are in different places they may have gas of widely different heat value. To overcome these difficulties, the author has made electric furnaces consisting of a single resistance coil of nichrome wire calibrated for a sufficient wattage to give a temperature of from 850 to 900°F. in a platinum crucible placed in its core. In 5 minutes after the electric current is applied to the coil, a constant and maximum temperature is reached and the covered platinum crucible is then inserted to be heated for 7 minutes. Weighings are made as with the regular method. Results show remarkably close agreement and are independent of variable gas supplies.

Porcelain crucibles should never be substituted for platinum crucibles for accurate work, as the porcelain requires a longer time to attain a constant temperature and therefore the duration of the application of the maximum temperature may not be the same as with one of platinum.

should become a powdery mass when combustion is complete. Combustion is assisted by inclining the crucible on the triangle during this test so as to admit air more freely for oxidation. After cooling in a desiccator, make a final weighing. The difference between this weight and that of the crucible without cover when empty is the ash. Weight of fixed carbon is obtained by subtracting the sum of the weights of moisture, volatile matter, and ash from the original weight of the sample of coal tested.

Weighings should be made with chemical scales sensitive to one-thousandth of the amount weighed. Two determinations of the complete proximate analysis should be made of each sample and the results should check within 0.5 per cent of the weight of the coal.

In a report, record, with the regular data, whether in the volatile test the coal coked into a single spongy mass or remained granular.

A.S.M.E. Methods.—Methods proposed for **proximate analysis** of coal by the Power Test Committee of the A.S.M.E. are those of the American Society for Testing Materials,¹ which are practically the same as for the American Chemical Society as given here.

To determine sulphur by Eschka's method, which is the one commonly used, heat 1 gram of coal mixed with 1 gram of magnesium oxide and $\frac{1}{2}$ gram of sodium carbonate for 1 hour, using an alcohol lamp. After cooling, mix with 1 gram of ammonium nitrate and heat the mixture 10 minutes; then dissolve in 200 cubic centimeters of water, heat and reduce by evaporation to 150 cubic centimeters, acidify with hydrochloric acid and filter. Add barium chloride to the filtrate and determine the sulphur by calculation from the quantity and composition of the barium thereby precipitated.

The carbon and the hydrogen are obtained by the use of the combustion apparatus. One-half gram of the pulverized air-dried coal is placed in a porcelain "boat," which is introduced between the copper roll of oxidized copper gauze and the copper oxide within the glass combustion tube. After the coal and the entire contents within have been thoroughly dried out by a sufficient preliminary heating, aided by a current of dry air, the furnace is set to work and the coal burned by first passing air and finally oxygen through the tube. The products of combustion are to be passed through potash bulbs and a chloride of calcium tube. The carbon dioxide produced by the combustion of the carbon is absorbed by the potash, and the water formed by the combustion of hydrogen, together with that due to the moisture in the air-dried coal, is taken up by the chloride of calcium. The quantity of carbon is determined by weighing the bulbs before and after, to obtain the weight of the carbon dioxide produced, and then calculating the weight of carbon from the known composition of the dioxide. Likewise, the quantity of hydrogen

¹ See A.S.T.M. Standards, 1921, pp. 765-767.

is determined by weighing the calcium tube before and after, which, after deducting the moisture in the air-dried coal, gives the amount of water produced and, dividing by 9, the amount of hydrogen.

The ultimate analysis of coal for sulphur, pure carbon, and hydrogen, as will be seen from the foregoing description, requires the use of so much chemical apparatus, and at best it is so complicated, that it is not likely to be done except in a fully equipped chemical laboratory. *It should not be undertaken by one who is not entirely familiar with all the details of the work.*

Hydrogen from Proximate Analysis.—Professor L. S. Marks¹ has given curves showing that the hydrogen in the coal which is not included in the moisture can be obtained accurately from the results of proximate analysis. Also it is then possible to obtain the chemical analysis of coal from the proximate analysis. This is important, as in order to make a complete analysis of a boiler it is necessary to find the percentage of total carbon in the coal and also the hydrogen in the volatile matter.

In the proximate analysis the combustible is equal to the sum of the volatile matter and fixed carbon, or it is equal to the coal as received less the ash and moisture.

The Marks' curves have been expressed by the following equations:

$$H = V \left(\frac{7.35}{V + 10} - 0.013 \right)$$

where in the foregoing and following equations,

V = the percentage by weight of volatile matter in combustible.

H = the percentage by weight of hydrogen in combustible.

C = the percentage by weight of volatile carbon in combustible.

N = the percentage by weight of nitrogen in combustible.

This equation has an accuracy for American coals of about ± 0.2 per cent.

$C = 0.02V^2$ for anthracite, and $C = 0.9(V - 10)$ for semianthracite.

$C = 0.9(V - 14)$ for bituminous and semibituminous.

$C = 0.9(V - 18)$ for lignites.

Sulphur in coal directly increases the value of V ; values of C calculated will be too high practically by the sulphur content of the combustible.

The nitrogen which comes off in the volatile matter can be obtained with an accuracy of ± 0.5 per cent by use of the following equations:

$N = 0.07V$ for anthracite and semianthracite,

$N = 2.10 - 0.012V$ for bituminous and lignite.

Purchasing Coal by Calorific Value (B.t.u.) and Analysis.—Every year there are more power plants purchasing their coal on the basis of analysis and calorific value. It may be generally assumed that shipments of coal coming from the same mine at different times will not vary

¹ *Power*, December, 1908.

a great deal in the composition of combustible. Moisture and ash may, however, vary considerably and every large shipment, preferably every car load, should be tested at least for moisture and ash content as a check to determine whether the specifications are being fulfilled. At frequent intervals, particularly when ash or moisture determinations indicate a doubtful quality, samples should be sent to some laboratory for a complete analysis. For only the moisture and ash determinations, no special equipment is needed. If moderately large weights are used, a fairly accurate "counter" or even a good platform scales may be used for weighing; and aside from this only a few sheet-metal or tin pans, one or two thermometers, and several Bunsen burners and porcelain crucibles are absolutely required. Besides the scales for weighing, the equipment needed can be purchased for \$10. For a drying box almost any kind of a discarded galvanized iron or tin vessel can be used. There should be a small hole in the top for the insertion of a thermometer.

To burn coal to ash with Bunsen burners requires the application of heat for a great many hours. Instead of providing air pressure for a blast lamp, some engineers prefer to buy oxygen in tanks¹ and burn the coal rapidly in a pure oxygen atmosphere.

A.S.T.M.² METHOD OF SAMPLING SOLID FUELS (ABRIDGED)

(APPROVED BY A.S.M.E., 1931)

For Solid-fuel Determinations (except Total Moisture)

Source of Sample.—The coal shall be sampled when it is being loaded into or unloaded from railroad cars, ships, barges, or wagons, or when discharged from supply bins, or from industrial railway cars, or grab buckets, or from any coal-conveying equipment, as the case may be. If the coal is crushed as received, samples usually can be taken advantageously after the coal has passed through the crusher. Samples collected from the surface of coal in piles or bins or in cars, ships, or barges are generally unreliable.³

¹ Linde Air Products Co., Buffalo, N. Y., with distributing stations in several large cities.

² American Society for Testing Materials.

³ Gross samples of the quantities designated must be taken, whether the coal to be sampled consists of a few tons or several hundred tons, because of the following cardinal principle in sampling coal that must be recognized and understood; that is, the effect of the chance inclusion or exclusion of too many or too few pieces of slate or other impurities in what, or from what, would otherwise have been a representative sample will cause the analysis to be in error accordingly, regardless of the tonnage sampled. For example, the chance inclusion or exclusion of 10 pounds too much or too little of impurities in or from an otherwise representative sample of 100 pounds would cause the analysis to show an error in ash content and in heat units of approximately 10 per cent, whereas for a 1,000-pound sample, the effect would be approximately only 1 per cent, being approximately the same whether the sample is collected from a 1-ton lot or from a lot consisting of several hundred tons.

Size of Increments in Sampling.—To collect samples, a shovel or specially designed tool,¹ or mechanical means should be used for taking equal portions or increments. For slack or small sizes of anthracite, increments as small as 5 to 10 pounds may be taken, but for run-of-mine or lump coal, increments should be at least 10 to 30 pounds.

Collection of Gross Sample.—The increments shall be regularly and systematically collected, so that the entire quantity of coal sampled will be represented proportionately in the gross sample, and with such frequency that a gross sample of the required amount shall be collected. The standard gross sample shall not be less than 1,000 pounds, except that for slack coal and small sizes of anthracite, in which the impurities do not exist in abnormal quantities or in pieces larger than $\frac{3}{4}$ inch, a gross sample of approximately 500 pounds shall be considered sufficient. If the coal contains an unusual amount of impurities, such as slate, and if the pieces of such impurities are very large, a gross sample of 1,500 pounds or more shall be collected. The gross sample should contain the same proportion of lump coal, fine coal, and impurities as is contained in the coal sampled. When coal is extremely lumpy, it is best to break a proportional amount of the lumps before taking the various increments of a sample. Provision should be made for the preservation of the "integrity" of the sample.

Crushing of Sample of Coal.—After the gross sample has been collected, it shall be systematically crushed, mixed, and reduced in quantity to a convenient size for transmittal to the laboratory. The sample may be crushed by hand or by any mechanical means, but under such conditions as shall prevent loss or the accidental admixture of foreign matter. Samples of the quantities indicated in Table XVII shall be crushed so that no pieces of coal and impurities will be greater in any dimension, as judged by eye, than specified for the sample before division into two approximately equal parts.

TABLE XVII.—WEIGHT AND SIZE OF SAMPLE

Weight of Sample to Be Divided, Pounds	Largest Size of Coal and Impuri- ties Allowable in Sample before Division, Inches
1,000 or over.....	1
500.....	$\frac{3}{4}$
250.....	$\frac{1}{2}$
125.....	$\frac{3}{8}$
60.....	$\frac{1}{4}$
30.....	$\frac{3}{16}$ to pass a No. 4 sieve

The progressive reduction in the weight of the sample to the quantities indicated in Table XVII shall be done by the following methods, which are illustrated in the *plates* in Fig. 238.

¹ For a more complete discussion of equipment for the sampling of coal as well as criticisms of sample specifications, see *Bull. 5*, The Pennsylvania Engineering Experiment Station, State College, Pa.

a. **Mixing and Reduction by Discarding Alternate Shovelfuls.**—The “alternate-shovel” method of reducing the gross sample shall be repeated until the sample is reduced to approximately 250 pounds and care shall be

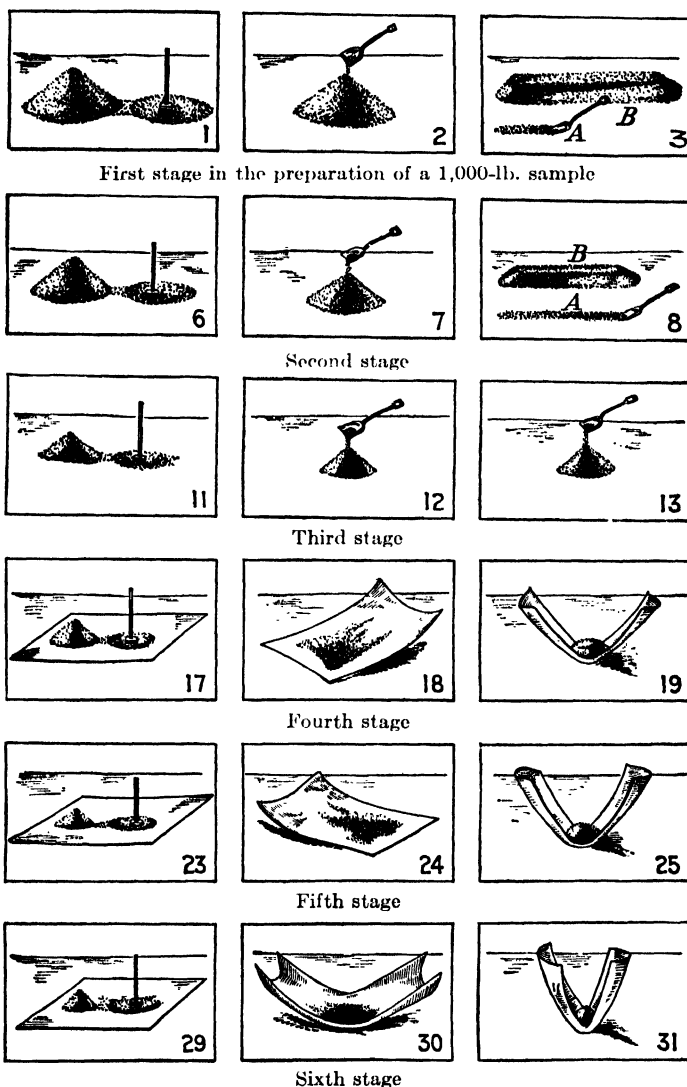
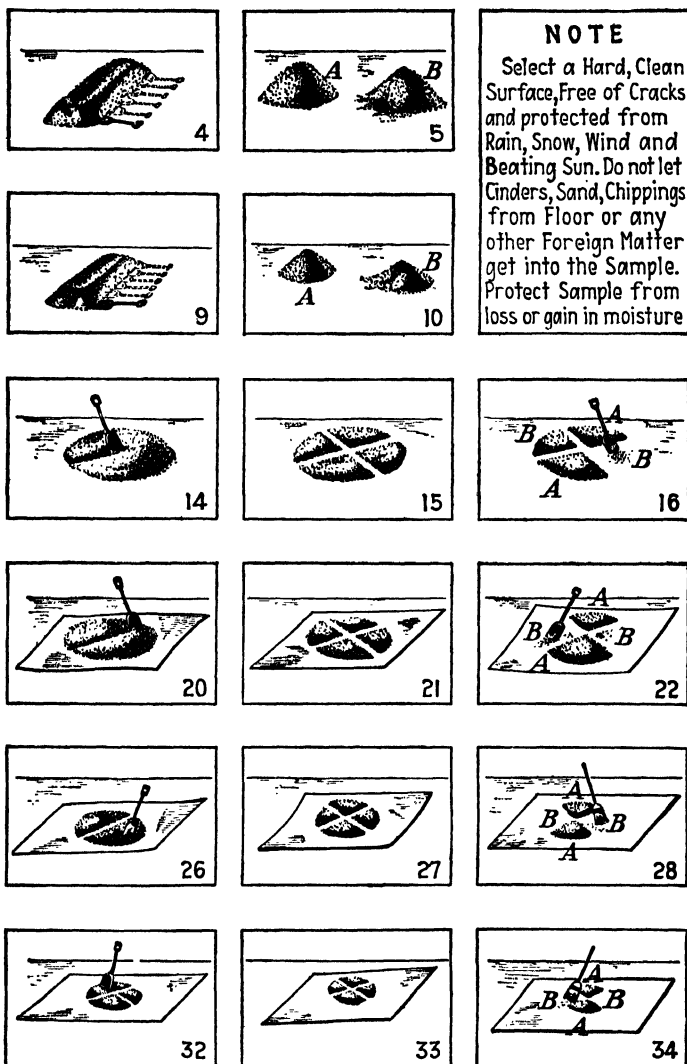


FIG. 238.—Bureau of Mines method

observed before each reduction in quantity that the sample has been crushed to the fineness prescribed in Table XVII.

The crushed coal shall be shoveled into a conical pile by depositing each shovelful of the sample coal on top of the preceding one and then formed into a long pile in the following manner: The sampler shall take a

shovelful of coal from the conical pile and spread it out in a straight line (Plate 3A or 8A) having a width equal to the width of the shovel and a length of 5 to 10 feet. His next shovelful shall be spread directly over the



of sampling solid fuel by quartering.

top of the first shovelful, but in the opposite direction, and so on, back and forth, the pile being occasionally flattened, until all the coal has been formed into one long pile (Plate 3B or 8B). The sampler shall then discard half of this pile, proceeding as follows:

Beginning at one side of the pile, at either end, and shoveling from the bottom of the pile, the sampler shall take one shovelful (shovelful 1, *Plate 4* or *9*) and set it aside; advancing along the side of the pile a distance equal to the width of the shovel, he shall take a second shovelful (shovelful 2) and discard it; again advancing in the same direction one shovel width, he shall take a third shovelful (shovelful 3) and add it to the first. The fourth (shovelful 4) shall be taken in a like manner and discarded, the fifth (shovelful 5) retained, and so on, the sampler advancing always in the same direction around the pile so that its size will be gradually reduced in a uniform manner. When the pile is removed, about half of the original quantity of coal should be contained in the new pile formed by the alternate shovelfuls which have been retained. (*Plate 5A* or *10A* shows the retained halves, and *Plate 5B* or *10B* the rejected halves.)

b. Mixing and Reduction by Quartering.—After the gross sample has been reduced by the foregoing method to approximately 250 pounds, further reduction in quantity shall be by the quartering method. Before each quartering, the sample shall be crushed to the fineness prescribed in *Fig. 238*.

Quantities of 125 to 250 pounds shall be thoroughly mixed by coning and reconing (*Plates 12* and *13*); quantities less than 125 pounds shall be placed on a suitable cloth, measuring about 6 by 8 feet, mixed by raising first one end of the cloth and then the other (*Plate 18, 24, or 30*), so as to roll the coal back and forth, and after being thoroughly mixed shall be formed into a conical pile by gathering together the four corners of the cloth (*Plate 19, 25, or 31*). The quartering of the conical pile shall be done as follows:

The cone shall be flattened, its apex being pressed vertically down with a shovel, or board, so that, after the pile has been quartered, each quarter will contain the material originally in it. The flattened mass, which shall be of uniform thickness and diameter shall then be marked into quarters (*Plate 14, 20, 26, or 32*) by two lines that intersect at right angles directly under a point corresponding to the apex of the original cone. The diagonally opposite quarters (*BB* in *Plate 16, 22, 28, or 34*) shall then be shoveled away and discarded and the space that they occupied brushed clean. The coal remaining shall be successively crushed, mixed, coned, and quartered until the sample is reduced to the desired quantity.

The laboratory sample shall be immediately divided into two parts, placed in suitable containers, and sealed in such a manner as to preclude tampering. One of these shall be sent to the laboratory for analysis, and the other retained at the plant until satisfactory analysis of the laboratory sample has been completed and reported.

In collecting and reducing *coke* samples, all instruments used for crushing shall be made of iron or steel, so that, in later preparation of the sample in the laboratory, particles mixed with the *coke* sample, due

to its abrasive action on crushing instruments, may be separated out by running the ground sample under a magnet.

Determination of Total Moisture

Collection of Moisture Sample of Solid Fuel.—The special sample for the *moisture* determination shall weigh approximately 100 pounds and shall be accumulated by placing in a waterproof receptacle, with a tight-fitting waterproof lid, small equal parts of freshly taken increments of the standard gross sample. The accumulated-moisture sample shall be rapidly crushed and reduced mechanically or by hand to about a 5-pound quantity, which shall be immediately placed in a container and sealed air-tight and forwarded to the laboratory without delay.

Only when equally representative results will be obtained with certainty shall the standard gross sample be used instead of the special moisture sample for the determination of total moisture.

Containers for Samples of Solid Fuels.—Samples in which the moisture content is important should always be shipped in moisture-tight containers. A galvanized-iron or tin can with an air-tight friction top or a screw top which is sealed with a rubber gasket and adhesive tape is best adapted to this purpose. Glass fruit jars with rubber gaskets may be used but require very careful packing to avoid breakage in transit. Obviously, samples in which the moisture content is of no importance need no special protection from loss of moisture.

Moisture Sample of Ash.—The moisture sample shall be taken from the collected gross sample of ash before the latter has been crushed for subdivision in preparation for the laboratory sample. A "grab" sample of from 10 to 20 pounds will serve as the moisture sample. Before taking this, mix the gross sample by turning it over several times with a shovel.

In taking this sample the least possible time should be used, in order to reduce the moisture loss by evaporation to a minimum. This moisture sample shall then be conveyed in a sealed container to the laboratory or place in which the moisture determination is to be made.

Moisture Determination of Ash.—The ash sample from the test shall be quickly crushed down in a jaw crusher to pass through a 4-mesh sieve. It shall now be reduced to approximately a 5-pound sample, weighed quickly, and spread over galvanized-iron pans. These pans containing the sample shall now be placed in an *air-drying* oven and shall have air passed over them at a temperature not exceeding 200°F. The process shall be continued until the weight loss is not more than 0.1 per cent per hour.

Determination of Combustible Matter.—*a.* Combustible in refuse shall be computed by means of difference in weight of total coal, computed weight of total dry ash from the ultimate analysis, and weight of boiler refuse, the heating value of the combustible to be taken as that of pure carbon.

b. For determination of the combustible in fuel siftings through the grate, the sifting sample shall be treated according to the method described under proximate analyses in A.S.T.M. Specification D22-21 for the determination of ash in coal.¹

¹ See A.S.T.M. Standards, 1921, p. 760.

The heating value of the combustible in the siftings shall be determined by the calorimetric method.

Test for Fusing Point of Ash.—At the time of writing this code there is no generally recognized method of testing for the fusibility of ash or of properly interpreting the results arrived at, as the fusing temperature varies considerably under different conditions, and for the same apparent fusing temperatures different results are obtained under different operating conditions. However, the method of test for the fusing temperature adopted by the American Society for Testing Materials (D22-21) has been tentatively approved.

POWDERED COAL

Gross Ash Sample from Furnace.—This shall be collected and weighed at convenient intervals, care being taken that none of the fine ash deposit is lost. If there are large slag formations, these should be broken down and weighed separately.

Preparing Laboratory Samples.—Laboratory samples from both the gross stack-ash sample¹ and the gross furnace-ash sample shall be prepared from said gross samples by the standard process of reducing, accompanied by quartering and mixing.² The laboratory samples shall weigh from 2 to 5 pounds. Separate samples shall be taken for the furnace and stack. If the slag is to be analyzed, the sample shall be prepared by crushing all of the slag so that it will pass through a $\frac{1}{8}$ -inch screen.

APPENDIX

FRAZER CLASSIFICATION FOR COAL

Anthracites.....	Volatile below 5 per cent
Semianthracites.....	Volatile 5-10 per cent
Semibituminous.....	Volatile 15-22 per cent
Bituminous, Eastern.....	{ Volatile 25-35 per cent Vein moist. 2-4 per cent
Bituminous, Mid-continental.....	
Black lignites or subbituminous.....	Volatile 35-45 per cent
	Vein moist. 17-20 per cent
Brown lignites.....	Volatile 25-45 per cent
	Vein moist. 20-25 per cent

QUANTITATIVE TEST FOR SOLIDS IN FLUE GASES

For the determination of solids such as cinders, an efficient cinder catcher through which the total gases are passed will lead to more accurate results than any sampling device which takes only a very small fraction of the total gases. Furthermore, owing to the relative slow movement of the gases in many natural-draft installations, it is

¹ Methods of collecting stack ash by cinder catchers are not reliable (A.S.M.E., Power Test Code on Solid Fuels). The more elaborate methods given in *Bull. 223*, U. S. Bur. Mines, are recommended.

² In cases of important acceptance tests, exportation of coal, etc., it is often imperative to have the analysis performed by such an organization as the Government laboratories afford. Under some conditions the Bureau of Mines will analyze and test coal.

impossible to obtain a representative sample by simple means through a small sampling tube. (There is insufficient kinetic energy in the particles to carry them through such tubes.) It is often impracticable to have the total gases passed through a cinder catcher, and for such cases no generally recognized method is in use.¹

¹ See report of 1915 of Chicago Association of Commerce Committee of Investigation on Smoke Abatement and Electrification of Railway Terminals.

CHAPTER IX

FLUE-GAS ANALYSIS

Flue-gas Analysis.—The analysis of flue gases in connection with tests of steam boilers gives a valuable means for determining the relative value of different methods of firing and of different types of furnaces. Errors in the analysis of flue gases are most often due to the inability to secure an average sample of the gases in the different parts of a flue or chimney. The composition is likely to vary considerably even during short intervals, and it is therefore desirable to adopt some method of sampling which will permit collecting the sample slowly and continuously for a considerable period.

A very simple and convenient sampling apparatus is shown in Fig. 239.

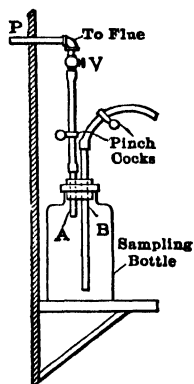


FIG. 239.—Gas sampling bottle.

The sample of gas is taken from the flue or chimney through the pipe *P* shown at the top of the figure. This pipe extends well into the flue and is preferably open-ended¹ as a better sample of gas can be taken than if the sampling pipe is slotted or is perforated along its length. The pipe outside the flue is connected by means of a short rubber tube to the sampling bottle. A valve *V* should be put as near as possible to the end of such a pipe, so that it can be closed up when the sampling bottle is removed for analysis. If the pipe cannot be closed, the suction in the flue will draw air into the pipe

and fill it so that when again connecting up the sampling bottle all this air must be removed before a true sample can be taken. The sampling bottle is preferably one with a wide neck, closed with a rubber stopper through which two glass tubes pass into the bottle, one reaching nearly to the bottom and the other entering only a little below the stopper. Tube *B* can be connected to an **aspirator** or **ejector**, or any similar device producing a suction, and there will be a steady flow of gas through the bottle. If still another short tube like *B* is put into the stopper, it can be used for the attachment of gas-analysis apparatus, making a very satisfactory arrangement, and it is then unnecessary to disconnect the sampling bottle from the pipe.

¹ For further discussion of sampling of flue gas, see A.S.M.E. flue-gas sampler (p. 268) and Test Code for Steam Generating Units—Flue-gas Analysis, p. 311.

Small aspirators or ejectors (Fig. 240) operating on the principle of an injector with a small stream of water which entrains the gases are very convenient for collecting samples continuously. A slightly different design made of pipe fittings is shown in Fig. 241. Water enters through a vertical nozzle *N* and in discharging as a jet at high velocity entrains air or gas drawn in through the side opening, and the mixture of atomized water and air is discharged with considerable velocity through the forcing tube *F* at the bottom.

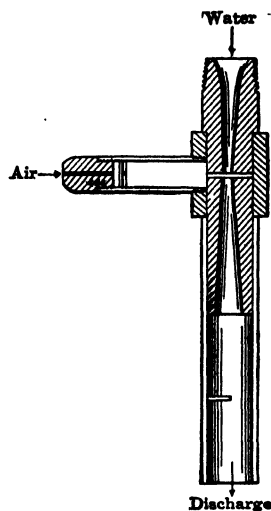


FIG. 240. — Water-jet aspirator or ejector.

If an aspirator is not available, the sampling bottle and the tube *B* may be filled with water. Then by gradually siphoning the water from the bottle the flue gases will be drawn in. By adjusting the valve *V* the rate of flow of the gases into the bottle can be regulated. The water will probably absorb some of the constituents of the gas. On this account very little water should be left in the bottle with the sample of gas. If the water is saturated with gases, as it will be from long use, this precaution is not so essential. Sampling and collecting bottles should have rubber rather than cork stoppers, because cork is too porous.¹

The sampling bottle and the tubes must be completely filled with the liquid before beginning to take the sample by the method of displacing water, because any air that is left in them will remain in the bottle and will be mixed with the sample of the gas. If the end of the stopper going into the bottle is made slightly conical, it will be easier to avoid entrapping bubbles of air at the top of the bottle when inserting the stopper.

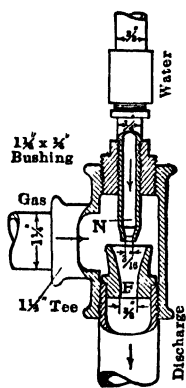


FIG. 241. — Ejector made of pipe fittings.

This type of sampling bottle when filled with liquid can be used also very conveniently by reversing the connections of its tubes, that is, by attaching the long tube to the pipe entering the flue, and then turning the bottle upside down. The liquid will then run out through the shorter tube and the gas will be drawn in to fill the bottle.

A portion of the gas can be removed from the sampling bottle into the measuring burette or tube required for making the analysis of the flue gas by connecting the short tube

¹ A good way to cut holes in rubber stoppers for tubes is with the ordinary drills used for metal work, using a drill considerably larger than the hole required.

of the sampling bottle to the burette or to some other part of the gas-analysis apparatus as may be required. If, now, the rubber tube connected to the longer tube of the sampling bottle when disconnected is put into a pail well filled with water, then, as the gas is withdrawn from the bottle, water will be drawn from the pail to displace it. One of the advantages of this sampling apparatus is that it can be easily made from the materials obtainable in almost any town or village. A 2-quart preserving jar with a rubber stopper to fit and tubes of glass, brass, or iron can be used to make up a very good apparatus. For accurate power-plant tests, these simple gas-collecting devices are not permitted, and more elaborate apparatus is required.

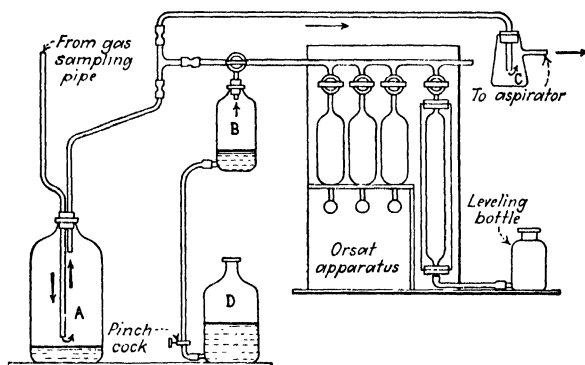


FIG. 242.—A.S.M.E. apparatus for flue-gas sampling.

Sampling Tube for Flue Gas (A.S.M.E. Design).—The standard flue-gas sampler shall consist of a single open-ended iron, copper, or glass exploring tube of $\frac{1}{4}$ to $\frac{3}{4}$ inch diameter and of which the *open ends* are placed at such positions that the collected sample has a composition representative of the average composition of the gas being sampled. A determination of such positions will require preliminary exploration and judgment. Pockets back of dampers and at bends shall be avoided.

Location of Flue-gas Temperature-measuring Instruments.—The thermometer bulb or hot junction or other device used for temperature measurements shall be placed as near the *open end of the sampling tube* as possible and in such a position that the stream flow impinges freely on the temperature-responsive elements, but, of course, not so as to interfere seriously with gas flow into sampling tube.

The thermometer or hot junction must be protected from radiation to colder or hotter surfaces, which may be in the vicinity, by providing metal, asbestos, or other shields so placed in the gas stream that no direct radiation can take place between the bulb or hot junction and the hot or cold surfaces.

Method of Collecting Flue-gas Sample.—The gas sample shall be carried from the sampling pipe (Fig. 242) in a single tube to the bottle A

for the separation of condensed moisture and tars, past the T-connection (the tube should be capillary bore) to one end of which the collecting bottle *B* is connected, through the suction bottle *C* half-filled with water, which indicates by bubbling that a sample is being drawn and the rate, and thence to the aspirator. The aspirating process shall be strong, continuous, and at a uniform rate. The collecting bottle is a standard 2-liter aspirating bottle fitted with a three-way stopcock (having a capillary bore) in the rubber stopper at the top. Any vessel *D* having a capacity somewhat greater than the bottle *B* may be used to contain the water used for displacement. It is connected to the bottle by a rubber tubing fitted with a pinchcock for regulating the flow of displacement water out of and into the bottle. With this apparatus a continuous sample¹ is drawn from the main stream of gases, a small part of this

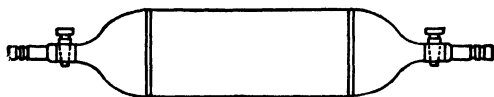


Fig. 243.—Container for flue gas.

sample is drawn off continuously for 15 or 30 minutes or other suitable period, and a portion of this sample is then analyzed in the Orsat apparatus. As only 2 to 3 minutes are required to fill the Orsat apparatus and empty the gas from the bottle, there is practically a continuous sample and analysis of the stream of gas aspirated from the breeching. If water is used in the bottle *B* and vessel *D*, it shall be saturated with the gas being analyzed; or concentrated brine may be used.

It is preferable to have the flue-gas sample drawn directly into the Orsat apparatus. When this is not feasible, the gas sample shall be collected at the aspirator in standard collecting bottles such as shown in Fig. 242 or in a tube of tin or glass such as illustrated by Fig. 243. *The gas samples may be collected in quantities proportioned to the rate of burning fuel, or they may be collected in equal quantities at equal intervals of time. In the latter case, the average analysis of the gases must be arrived at by weighing the separate determinations proportionally to the weights of fuel burned during the times of collecting the samples from which the analyses are made.* Whenever possible, the analysis shall be made immediately after collection of the sample.

Flue-gas Analysis.—The gas samples collected in the manner described above shall be analyzed for carbon dioxide, carbon monoxide, and oxygen by means of an Orsat or other similar apparatus (see pages 272 and 275). When more than 0.5 per cent of carbon monoxide is present, the carbon monoxide shall be removed with two cuprous chloride

¹ When the gas is at a pressure below normal, it may be drawn in by an aspirator. When, however, it is under pressure above normal, it will readily displace the water in the container.

pipettes in the Orsat apparatus. If hydrogen is present in the products, an apparatus such as the Hempel (page 276) or other suitable apparatus should be used.

Since there is nearly always a great variation in the composition of the gases in the various parts of a flue or chimney, it is not very likely that a pipe having a long slot will give a "fair" sample. Obviously,

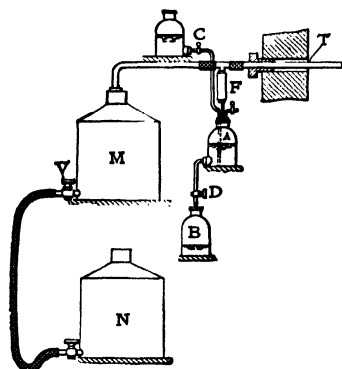


FIG. 244.—English gas sampling apparatus.

most of the gas will enter the slot in that portion of its length nearest the collecting apparatus. Another device often used for a sampling tube consists of a horizontal pipe into which a number of branch tubes are fitted. These branch tubes are arranged so that the openings at their ends will take samples from the different parts of the flue or chimney in which they are placed. The A.S.M.E. Test Code for flue-gas sampling requires the use of a single open-ended tube (page 268).

Figure 244 shows a sampling device used extensively in England. It consists of an iron tube *T* of $\frac{3}{4}$ -inch iron pipe open only at the end in the flue. The end is located carefully, however, in English practice so that it lies well into the current of the gases. The pipe *T* is connected to a vessel *M* about 2 feet in diameter. The bottom is connected to another vessel of about the same size which is open at the top to the atmosphere. Samples for analysis are taken from the small bottle *A*. Vessels *M* and *N* are provided to prevent stagnation of the gases in the pipes. Rubber tubing is used in only very short lengths because it is said to be *more or less porous to carbon dioxide*. When the test is started, the bottle *A* is full of mercury, cock *C* is closed, and *D* is open just enough to permit a slow flow of mercury into *B*. When a new test is to be started, the mercury in *B* is poured into *C*, and *A* is filled and slowly emptied as before.

When a sample of flue gas is taken from the flue at a considerable distance from the furnace, it is likely to become mixed with air leaking through the brickwork of the boiler setting, and the analysis will not show the true relations between the volumes of the so-called flue gases and the excess of air. To prevent as much as possible this leakage of air, the joints in the masonry must be examined and repaired if necessary and the samples must be taken as near as possible to the fire. It must be borne in mind, however, that they must be drawn very slowly from the hot flue in order that they will be cooled down gradually to avoid dissociation. If dissociation occurs in the sample, the analysis may show results entirely different from the true composition of the gas in the flue. It is also

difficult to prevent the entrance of air into a flue through the bearings of dampers, and whenever it is possible the sample of flue gas should be obtained between the furnace and the damper. At high temperatures sampling tubes of other metals than platinum¹ or nickel are not quite satisfactory, since by their oxidation they abstract oxygen from the gases passing through them.

Apparatus for the Analysis of Flue Gases.—Samples of flue gases contain in varying amounts carbon dioxide (carbonic acid), oxygen, carbon monoxide, nitrogen, unburned hydrocarbons, and occasionally some free hydrogen. For the data which an engineer usually requires it is not necessary to determine by direct analysis more than three of these: carbon dioxide, CO₂, oxygen, O₂, and carbon monoxide, CO.

The determination of carbon monoxide with the facilities and the portable apparatus ordinarily available in engineering laboratories is often somewhat doubtful. Some authorities state that there is rarely more than a trace of carbon monoxide to be found in the gases from combustion in the ordinary types of furnaces. When more than 1 per cent carbon monoxide is shown by the analysis and the carbon dioxide determination is not over 14 per cent, it may usually be assumed that a large part of what is taken to be carbon monoxide is oxygen which was not absorbed by the proper reagent.

In Table XVII A a set of typical analyses of flue gases is shown. The determinations were made by Scheurer-Kestner with coal from Ronchamp. Other analyses of flue gases may be checked by a comparison with this table. Thus when the analysis shows from 8 to 12 per cent carbon dioxide, the sum of the percentages of carbon dioxide and oxygen will probably be between 19 and 20.

TABLE XVII A.—PERCENTAGE COMPOSITION OF FLUE GAS

CO ₂	O ₂	CO	N	Hydrocarbons
8.2	11.3	0.2	79.8	0.5
10.8	9.0	0.2	79.7	0.3
12.9	5.5	0.2	80.3	1.1
13.4	4.4	0.2	80.2	1.8
14.6	2.8	0.3	80.6	1.7

In the portable apparatus used by engineers for the analysis of flue gases a separate pipette or treating tube is provided for each reagent, and the chemicals used are of greater strength than the reagents used by some chemists. The following reagents give satisfactory results in a portable apparatus:

¹ Porcelain and annealed glass are also satisfactory materials for making sampling tubes for very hot flues.

1. For absorbing carbon dioxide, a solution of one part of potassic hydrate or caustic potash, KOH, dissolved in two parts by weight of water is generally used.

2. For absorbing oxygen, either an alkaline solution of pyrogallie acid¹ or sticks of phosphorus are employed.

The alkaline solution of pyrogallie acid is prepared by mixing together preferably in the absorption pipette or treating tube, to prevent access of air, 5 grams of pyrogallie acid powder and 100 cubic centimeters of potassic hydrate, KOH, solution prepared as explained above.

To make the absorption more rapid, some engineers use a solution very much stronger in pyrogallie. This is not good practice, as stronger solutions are likely to evolve carbon monoxide in the presence of oxygen.

Phosphorus is more rapid in its action than the pyrogallate but has the disadvantage of being difficult to use, as it must be handled under water.

3. For absorbing carbon monoxide, a hydrochloric acid solution of cuprous chloride is used. This is prepared by dissolving about 10 grams of cupric oxide in from 100 to 200 cubic centimeters of concentrated hydrochloric acid. This solution must be allowed to remain in a bottle tightly closed and well filled with copper wire or gauze, until the cupric chloride is reduced to cuprous chloride. In this latter state the liquid will be colorless. Exposure to the air produces a brown color, indicating the cupric state.

After a time these reagents must be replaced by new solutions. The potassic hydrate solution may be used until each volume has absorbed 40 volumes of carbon dioxide. Pyrogallie acid solution deteriorates rapidly and each volume should be expected to absorb only one or two volumes of oxygen. Cuprous chloride will absorb an equal volume of carbon monoxide.

Portable devices for the analysis of flue gases are generally known as "Orsat" apparatus. Of these there are various types. The one devised by **Fisher**, shown in Fig. 245, has been used extensively. It consists of a measuring tube *M*, surrounded by a water jacket, and a set of absorption pipettes *A*, *B*, *C*, each filled with a reagent. Each of these pipettes (Fig. 246) consists of two glass vessels connected by a U-shaped glass tube at the bottom. One end of these pipettes is joined by means of a short piece of rubber tubing to a glass yoke *T*, which is designed for attachment at one end to a tube leading to the sampling bottle and at the other end to the measuring tube. A water bottle *W* is connected by a flexible rubber tube *P* to the bottom of the measuring tube.

Before a sample of gas is taken into the apparatus for analysis, certain adjustments must be made. In the first place, the reagents in the

¹ When the temperature is lower than about 55°F., this reagent does not give satisfactory results.

pipettes must all be brought to a standard level at some arbitrary point, usually indicated by a scratch on the glass tube just below the short rubber tube connecting it to the manifold yoke. This adjustment is accomplished by opening, one at a time, the valves at the tops of the pipette and removing the air (or the gas, as the case may be) from it by lowering the water level in the measuring tube. The position of the water bottle determines, of course, the level of the water in the measuring tube. When all the air and gases remaining from a previous test have

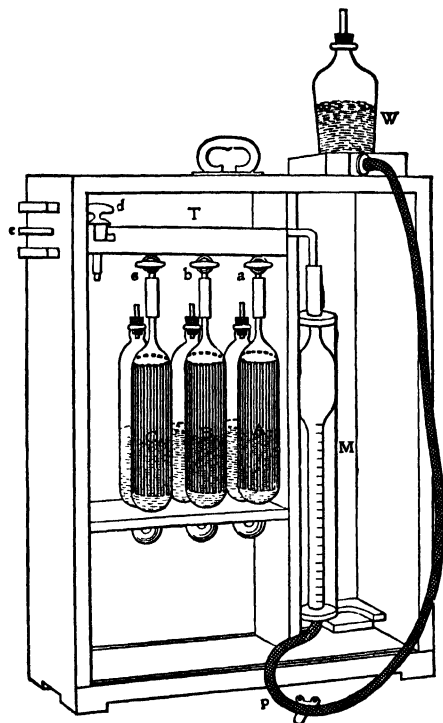


FIG. 245.—Fischer's "Orsat" apparatus.

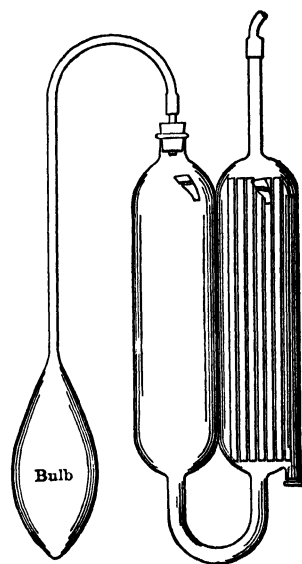


FIG. 246.—Pipette of Fischer's "Orsat" apparatus.

been expelled from the apparatus by filling the measuring tube and the tubes comprising the yoke with water, one of the tubes in the sampling bottle should be connected to the apparatus at *e* and by opening the valve in the yoke at that end and lowering gradually the level of the water in the measuring tube *M* a sample is obtained for analysis. This sample must be measured by the scale on the measuring tube at atmospheric pressure¹ to the nearest tenth of a cubic centimeter.² After

¹ The pressure of the gas is "atmospheric" when the water bottle is held so that the water in it and that in the measuring tube are at the same level.

² The scales of practically all measuring tubes used for gas-analysis apparatus are graduated in cubic centimeters.

the measurement has been made and recorded, if the cock in the tube leading to the absorption pipette containing the reagent for absorbing carbon dioxide is opened and then the water bottle is raised, all of the measured sample of gas can be forced over into the pipette. The reagent acts more rapidly on the gas if the water bottle is raised and lowered a few times. This movement of the water in the measuring tube agitates the gas and also the reagent and exposes more of the gas to the direct action of the absorbent. To increase the surface over which the reagents can act, the pipettes are filled with small glass tubes. When the gas has been in the first pipette for about a minute, it should be drawn back into the measuring tube with the level of the reagent brought back to the mark where it was originally, and the cock should be closed. The pressure of the gas is again made atmospheric and its volume measured. Now repeat this operation until **two or three measurements are obtained which are alike**, showing that all the carbon dioxide has been absorbed. Then the cock on the tube leading to the pipette containing the absorbent for oxygen can be opened, the gas forced over, and measured several times until a constant volume is observed. Finally the gas is passed into the third pipette for absorbing carbon monoxide, repeating the operation of measuring as with the other pipettes.

The absorption of oxygen will usually require considerably more time than the determinations of the carbonic acid, CO_2 , and the carbon monoxide, CO , so that it is unnecessary to measure the gas until it has been exposed for about 3 minutes to the reagent. Soft-rubber bulbs or bags (see Fig. 246) should be attached by means of glass tubes to the corks shown in the pipettes on the farther side in Fig. 245 and are provided to protect the reagents from absorbing oxygen from the air. Both pyrogalllic acid and cuprous chloride will absorb oxygen from atmospheric air, so that the access of fresh air must be prevented. The rubber bags are useful also for producing alternately, with the pressure of the hand, suction and pressure for agitating the reagents.

Allen-Moyer Gas Apparatus.—A form of flue-gas apparatus particularly suitable for portable use, in which renewals of broken parts can be cheaply and easily made, is illustrated in Fig. 247. This apparatus, designed by the late Prof. John R. Allen and the author,¹ is also suitable for the use of engineers because the pipettes containing the reagents can be removed from the apparatus very easily for changing solutions. They can be emptied, refilled, and replaced in a very short time. The absorption pipettes (Fig. 248) are made simply of two glass test tubes, the smaller one inside the larger one. The small test tube is held inverted and has a very small glass "capillary" tube fused into its closed end. The outer tube is closed at the top by a rubber stopper through which the capillary portion of the inner tube passes. Very small glass tubes are

¹ Made by Bausch & Lomb Co., Rochester, N. Y.

placed in the inner test tube to increase the surface for the action of the reagent. The complete pipette is held in place by means of a hard-rubber disk supported on brass screws. The level of the reagent in the pipette is established when the air in the inner tube is drawn out and the level of the liquid rises to a mark on the glass capillary tube. In the usual forms of the Orsat apparatus the pipettes invariably become leaky at the stopper provided for emptying. In the Allen-Moyer apparatus there is no opportunity for such leakage.

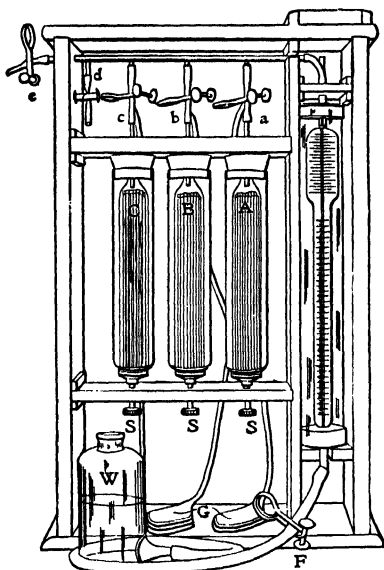


FIG. 247.—Allen-Moyer gas apparatus.



FIG. 248.—Absorption pipette of Allen-Moyer gas apparatus.

When the sample of the gas is passed through the capillary tube into the inner test tube, the reagent is displaced and raises the level in the outer test tube. Similarly, when the gas is passed back into the measuring tube, the level falls in the outer tube, rises in the inner one, and is brought back to the original level at the mark on the capillary tube. Otherwise the method of operation is the same as described for Fischer's apparatus (Fig. 245).

In this apparatus the measuring tube *M* and water bottle *W* are of the conventional type. The yoke is also similar, although usually made of hard rubber to avoid breaking it in transportation. It has also spring pinchcocks instead of ground-glass cocks. When glass cocks are used by inexperienced persons, all sorts of difficulties are likely to result, as it often happens that they are not pressed into their seats tightly enough to prevent the loss of gas or the entrance of air. Sometimes the glass cocks will be put into their seats so tightly that it is impossible to move

them without breaking. These difficulties, although met often enough in laboratory work, are still more frequently observed in practice.

It sometimes happens that, when *a*, *b*, or *c* is open and the pinchcock on the tube between *M* and *W* is closed, the reagent in *A*, *B*, or *C* falls, owing not to a leak, as is usually supposed, but to the weight of the column of the reagent expanding the gas.

In case any of the reagent in *A* or *B* is drawn over into the measuring tube and into the water, the analysis is not spoiled but may be continued by flushing out the tubes with water through *d* or *e*, or the addition of a little hydrochloric acid to the water in *W* will neutralize the hydrate or pyrogallate and the washing may be postponed until convenient.

To remove pipette *A*, *B*, or *C* when necessary to renew the reagents, disconnect the gas bags and the rubber tube which connects the glass capillary and rubber capillary tubes, loosen the supporting screw, and lift the pipette out. The

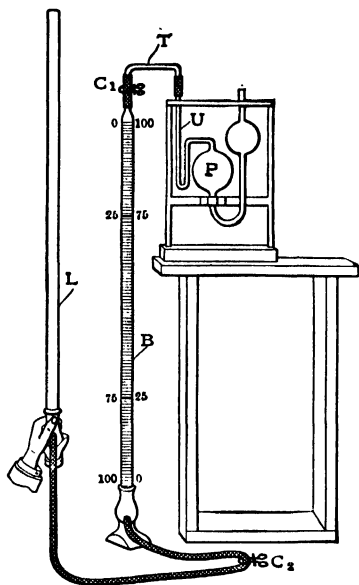


FIG. 249.—Hempel gas apparatus.

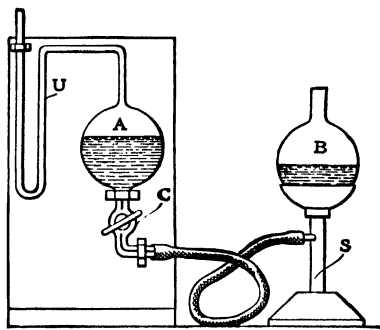


FIG. 250.—Explosion pipette.

rubber stopper may now be removed and solutions changed.

Gases should be cooled well in the sampling bottle before beginning the analysis, because such gases change $\frac{1}{49}$ of their volume for a variation of 1°F. , or a change of 1 per cent in volume for 4.91° . As an example, if the actual percentage of carbon dioxide is 10, and during the time required for analysis the temperature changed 4.91°F. , then there will have been a shrinkage of a volume of 1 per cent due to temperature, and the apparent volume of carbon dioxide will be 11 instead of 10.

Producer-gas Analysis.—For the analysis of producer and city illuminating gases, which are more complex than the flue gases from coal furnaces, the **Hempel apparatus** is generally used. Typical parts of the apparatus are shown in Figs. 249 and 250. It is slightly more difficult to operate and must be handled with greater care than a simpler portable apparatus arranged for flue-gas analysis.

Essential parts of the apparatus are shown in Fig. 249. They are the leveling tube *L*, a measuring burette *B*, and an absorption pipette *P*. In the operation of the apparatus the pinchcocks *C*₁ and *C*₂ are opened and water which has been thoroughly saturated with the kind of gas to be analyzed is poured into the leveling tube until both tubes are about half full. Now raise the leveling tube *L* so that the water from it flows into the burette *B*, making it entirely full. The pinchcock *C*₁ should then be closed and the rubber capillary tubing connected at the pinchcock to the pipe from which the gas for analysis is to be taken. After connecting to the gas pipe, again open the pinchcock *C*₁ and draw a little more than 100 cubic centimeters of gas into the burette. Allow the apparatus to stand a minute in this position to permit the water clinging to the sides of the burette to drain. Now close the pinchcock *C*₁ and by raising the leveling tube compress the gas in the burette until the meniscus stands at the 100 cubic centimeter mark, close the pinchcock *C*₂ on the lower length of rubber tubing, and open the pinchcock *C*₁ at the top of the burette momentarily to release the pressure in it. During this adjustment hold the leveling tube so that the surface of the water in it is at the same level as in the burette. There will be exactly 100 cubic centimeters of gas in the burette at atmospheric pressure if, when the pinchcock *C*₁ is opened, the meniscus remains at the 100 mark. If, however, the meniscus shifts from the 100 mark, the adjustment will have to be repeated.

Constituents of the gas must be absorbed in the following order: (1) carbon dioxide, CO₂, with potassium hydrate, KOH,¹ or sodium hydrate, NaOH; (2) "illuminant hydrocarbons" (ethylene C₂H₂, and benzene C₆H₆ in combination) with saturated bromine water² or fuming sulphuric acid; (3) oxygen, O₂, with caustic pyrogallous acid; (4) carbon monoxide, CO, with cuprous chloride; (5) marsh gas or methane, CH₄, and hydrogen, H₂.

A pipette must be provided for each of the reagents and those for pyrogallous acid and cuprous chloride which absorb oxygen must be provided with water seals to keep out air. Both ends of the pipette for fuming sulphuric acid must be kept closed except when an absorption is being made.

After the sample has been collected in the burette, the latter is attached to an absorption pipette as shown in Fig. 249 by a short piece of bent glass capillary tubing. Before this attachment is made, the glass capillary should be filled with water by means of a medicine dropper, so as to avoid the error of entrapping air in this tubing. The pinchcock *C*₁ should now be open and by raising the leveling tube all the gas

¹ These reagents are the same as used in the flue-gas apparatus.

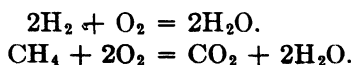
² Power Test Committee of A.S.M.E. specified bromine water for the hydrocarbons and sodium hydrate for carbon dioxide. Sodium hydrate is cheaper than potassium hydrate but the latter is the more rapid absorbent.

should be forced over into the pipette until the water from the burette fills the glass capillary connecting tube. Now close the pinchcock C_1 and shake the pipette lightly to give the gas the best sort of contact with the absorption reagent. After being shaken for 2 or 3 minutes, the gas should be drawn back into the burette by lowering the leveling tube until the solution from the pipette fills the connecting capillary tube, when the pinchcock C_1 should be closed. The surfaces of the water in the leveling tube and in the burette should now be brought to the same level. The reading on the scale observed after draining the burette for 2 minutes as read at the bottom of meniscus should be recorded. The process must be repeated as in the operation of an apparatus for flue-gas analysis until the reading is constant. In this way by the use of the proper pipettes all the constituents of the gas to be analyzed, with the exception of hydrogen and methane, are determined.

Explosion Tests.—For determining hydrogen and marsh gas, CH_4 , combustion tests are made in a pipette like Fig. 250, which is made particularly strong for exploding gases over mercury. In the upper portion of the pipette A there are two very fine platinum wires which are fused into the glass from opposite sides. When the spark from an induction coil is made to pass through the air gap between the two wires, the explosive mixture contained is ignited. The explosion takes place with considerable force so that extreme precautions should be taken to observe that the glass cock C and a very strong screw type of pinchcock at the end of the capillary tube U are very firmly closed. It is also very desirable to hold a wire screen about a foot square between the operator and the pipette when the spark is made and the explosion occurs.

Procedure for the explosion pipette is as follows: Measure about 15 cubic centimeters (call this m) of the n cubic centimeters of gas remaining after the absorption of carbon monoxide by the cuprous chloride and put the gas remaining in the burette into the pipette for absorbing carbon monoxide. Now transfer these 15 cubic centimeters to the explosion pipette and immediately thereafter measure about 85 cubic centimeters of air and add this to the gas in the explosion pipette. Then after closing both cocks very carefully and tightly and shaking lightly to insure a good mixture, close the electric circuit momentarily through the induction coil to cause the explosion. After the explosion, and allowing a little time for cooling, measure the gas in the burette. Then pass the same gas into the carbon dioxide pipette, measure the absorption of carbon dioxide in the burette, call this V cubic centimeters, and finally pass the remainder into the oxygen pipette and determine the volume of oxygen remaining.

In the explosion the following reactions took place:



If y is the number of cubic centimeters of oxygen absorbed after the explosion, then since there is 20.8 per cent of oxygen by volume in air, and if x cubic centimeters of air were used to make the explosive mixture, the oxygen supplied is $0.208 x$, and the oxygen used in the explosion is $0.208 x - y$. If it is assumed that the water is all condensed and has practically no volume, the equations above show, as regards volumes,

$$\text{Contraction of volume of gas } (z) = \frac{3}{2}\text{H}_2 + 2\text{CH}_4. \quad (64)$$

$$\text{Oxygen used } (0.208 x - y) = \frac{1}{2}\text{H}_2 + 2\text{CH}_4. \quad (65)$$

$$\text{CO}_2 \text{ formed } (v) = \text{CH}_4. \quad (66)$$

Subtracting Eq. (65) from Eq. (64),

$$z - [0.208 x - y] = \text{H}_2 \text{ (in } m \text{ cubic centimeters of "residual" gas),}$$

$$\text{and total H}_2 \text{ in the original sample} = \frac{n}{m} \{z - [0.208 x - y]\}.$$

Similarly, methane in m cubic centimeters of "residual" gas is the absorption of carbon dioxide (v) and the total methane in the original sample is $\frac{nv}{m}$.

Volume of nitrogen, N, can then be obtained by subtracting the sum of all the constituents now determined from the original volume of the sample of gas. Calculation of nitrogen content is, however, a good check on the accuracy of the analysis. Thus if, as above, x cubic centimeters of air were mixed with the m cubic centimeters of "residual" gas for the explosion test, then $0.208 x$ was oxygen and $(x - 0.208 x)$ was nitrogen. After the carbon dioxide and oxygen were absorbed from the products of the explosion, there were, say, w cubic centimeters of nitrogen remaining, so that the m cubic centimeters of "residual" gas contained $w - (x - 0.208 x)$ of nitrogen. Nitrogen content of the original sample is therefore $\frac{n}{m}[w - x(1 - 0.208)]$.

Results of the analysis should be tabulated and the accuracy as regards the "nitrogen check" stated clearly.

Coefficient of Dilution.—The coefficient of dilution is the ratio of the volume of the air supplied to the volume theoretically necessary to provide the oxygen required for combustion. It will now be shown how this coefficient can be calculated from an analysis of the flue gases.

Oxygen, when combining with carbon to form carbon dioxide, produces a volume equal to itself; thus



and in forming carbon monoxide produces twice the volume



Now if we use symbols to designate the percentages by volume of the gases in a sample of flue gas as follows:

a is the percentage by volume CO_2 .

b is the percentage by volume O_2 .

c is the percentage by volume CO .

d is the percentage by volume N_2 (nitrogen).

Then the volume occupied by the free oxygen in the air before combining with the carbon is $a + b + \frac{1}{2}c$ per cent, while that required for **complete** combustion is obviously $a + c$ per cent.

The coefficient of dilution¹ is therefore,

$$\frac{a + b + \frac{1}{2}c}{a + c}. \quad (69)$$

In a little different form the reactions given in the last paragraph may be stated (1) for carbon burned to carbon dioxide,



and (2) for carbon burned to carbon monoxide,



¹ This coefficient is variously defined so that in stating a result the method of computation should be given. In the use of this formula, it is assumed that all the oxygen necessary for combustion is used for combination with carbon, no allowance being made for that required for the combustion of the hydrogen in the fuel. Many engineers write it thus:

$$x = \frac{a + b + \frac{1}{2}c}{a + \frac{1}{2}c},$$

where the denominator as given is the air required for the kind of combustion as indicated by the analysis.

Another method sometimes used is based on the nitrogen content. Practically all the nitrogen indicated by the analysis is due to the total air supplied, and using d for the percentage of nitrogen we can write for the nitrogen which was a part of that used for combustion, approximately,

$$d - \frac{7}{11}(b - \frac{1}{2}c),$$

and

$$x = \frac{d}{d - \frac{7}{11}(b - \frac{1}{2}c)}.$$

Professor R. L. Daugherty, however, states that this formula gives very accurate results in most cases, *especially for oil fuels*. A simplified variation of the last formula, assuming complete combustion of carbon to carbon dioxide, no hydrogen in the fuel, and that the air supplied contains 20.8 per cent by volume of oxygen, is stated thus:

$$x = \frac{20.8}{20.8 - b}.$$

The ratio of the theoretical volume of the carbon burned to carbon dioxide to the volume burned to carbon monoxide is the same as the ratio of the volume of carbon dioxide in the products of combustion to the volume of carbon monoxide. Further, since the ratio of volumes of the carbon **vapor** is obviously the same as the ratio of the corresponding weights, we may say that in a mixture of gases the ratio of the **weight** of carbon required to produce the carbon dioxide to the **weight** needed for the carbon monoxide is equal to the ratio of the **volume** of carbon dioxide in the mixture to the **volume** of carbon monoxide.

The atomic weights of carbon and oxygen show [Eq. (70)] that the volume of oxygen is $2\frac{2}{3}(\frac{8}{4})$ times as heavy as an equal volume of carbon vapor. It follows then that for burning 1 pound of carbon to carbon dioxide, $2\frac{2}{3}$ pounds of oxygen are required. The other reaction [Eq. (71)] shows that $1\frac{1}{3}$ pounds of oxygen are required to burn 1 pound of carbon to carbon monoxide.

In the general case we are considering and using symbols a and c , respectively, as before to represent the volumes of carbon dioxide and of carbon monoxide in the flue gases, then the weight of the carbon burned to carbon dioxide is to the weight burned to carbon monoxide as a is to c , and if $\frac{a}{a+c}$ represents the weight of carbon burned to carbon dioxide and $\frac{c}{a+c}$ the weight burned to carbon monoxide, then the weight of oxygen required per pound of carbon is $2\frac{2}{3}\left(\frac{a}{a+c}\right) + 1\frac{1}{3}\left(\frac{c}{a+c}\right)$ and the weight of **air**¹ per pound of carbon is, in pounds,

$$\frac{100}{23}\left[2\frac{2}{3}\left(\frac{a}{a+c}\right) + 1\frac{1}{3}\left(\frac{c}{a+c}\right)\right]. \quad (72)$$

If z is the percentage by weight of carbon in the coal, then the weight of **air in pounds per pound of coal** is

$$\frac{100z}{23}\left[2\frac{2}{3}\left(\frac{a}{a+c}\right) + 1\frac{1}{3}\left(\frac{c}{a+c}\right)\right]. \quad (73)$$

Volumetric analyses of the flue gases can be used also to calculate the weight of the products of combustion (flue gases) per pound of coal burned and also the heat units lost in these gases. For this calculation the relations of the molecular weights are important.

Molecular weight of $\text{CO}_2 = 44$; $\text{O}_2 = 32$; $\text{CO} = 28$; $\text{N}_2 = 28$; $\text{C} = 12$.

¹ Weight of air may be checked with Hale's and Jacobus' Eqs. (76), (77), and (78), pp. 317, 330; also by the approximate method that $\frac{1}{4}$ pound of air needed per 1000 B.t.u. in heating value of coal or oil fuel.

Now in a sample of x pounds of flue gases in which the percentages by volume are represented by the symbols a, b, c, d , the relative percentages by weights of the constituents will be

$$\text{CO}_2 = \frac{44a}{x}; \text{O}_2 = \frac{32b}{x}; \text{CO} = \frac{28c}{x}; \text{N}_2 = \frac{28d}{x}; \text{ and we can write further}$$

$$\text{Weight of carbon burned to CO}_2 \text{ in } x \text{ pounds of gas} = \frac{11}{4} \times 44a = 12a.$$

$$\text{Weight of carbon burned to CO in } x \text{ pounds of gas} = \frac{11}{4} \times 28c = 12c.$$

$$\text{Total weight of carbon burned in } x \text{ pounds of gas} = 12(a + c).$$

$$\text{Total weight of carbon burned per pound of gas} = 12 \frac{(a + c)}{x}.$$

$$\text{Total weight of gas generated per pound of carbon} = \frac{x}{12(a + c)}.$$

Of this total weight of gas as expressed by the last equation the constituents are distributed in percentages by weight as follows; since the weight of carbon dioxide in samples per pound of carbon burned is,

$$\frac{44ax}{12x(a + c)}.$$

We may then write

$$\text{CO}_2 = w_1 = \frac{44a}{12(a + c)} \text{ pounds;}$$

and, similarly,

$$\text{O}_2 = w_2 = \frac{32b}{12(a + c)} \text{ pounds;}$$

$$\text{CO} = w_3 = \frac{28c}{12(a + c)} \text{ pounds;}$$

$$\text{N}_2 = w_4 = \frac{28d}{12(a + c)} \text{ pounds.}$$

Total weight of gases w_g per pound of coal burned, if there is z per cent¹ of carbon in the coal, is, in pounds,

$$w_g = \frac{z(44a + 32b + 28c + 28d)}{12(a + c)100}. \quad (74)$$

Now if we represent by t_f and t_a the temperatures, respectively, in degrees Fahrenheit of the gases in the flue and of the air entering the furnace, then the heat lost in the flue gases Q_g per pound of coal is, if values of specific heats are inserted,²

$$Q_g = \frac{z}{100}(0.217w_1 + 0.217w_2 + 0.245w_3 + 0.244w_4)(t_f - t_a).$$

¹ It may be assumed for very approximate values that $z = 1 - y$ where y is the percentage of ash and moisture in the coal.

² This method of finding the heat escaping in the flue gases may be used to correct determinations made with the Junkers calorimeter (p. 239) when the products of combustion are discharged at a temperature different from that of the room.

The total heat generated Q_0 by the more or less incomplete combustion of 1 pound of coal when there are a and c percentages by volume, respectively, of carbon dioxide and carbon monoxide in the flue gas is

$$Q_0 = \frac{z}{100} \left(\frac{a}{a+c} \times 14,600 + \frac{c}{a+c} \times 4,400 \right) \text{ in B.t.u.,} \quad (75)$$

since the heat of combustion of carbon when burned to carbon dioxide is approximately 14,600 and when burned to carbon monoxide is about 4400 B.t.u.

Finally, if Q_p is the heat from perfect combustion or the "calorific" value in B.t.u. of a pound of coal, then the **efficiency of the furnace**¹ = $\frac{Q_0}{Q_p}$. The percentage of heat from perfect combustion lost in the flue gases = $\frac{Q_g}{Q_p}$.

Recording Apparatus for Determining Carbon Dioxide.—Several kinds of apparatus for making a continuous record of the percentage by volume of carbon dioxide in flue gases depend on utilization of its change in absorption by potassium hydrate. Another kind depends upon the change of *pressure* of a stream of gas flowing through two apertures, when the carbon dioxide is absorbed. This is the method shown diagrammatically in Fig. 251. The gas to be analyzed is drawn through the two apertures at A and B by a constant suction produced by an aspirator.

If these apertures are kept at the same temperature, the suction or partial vacuum in the chamber between the two apertures will remain constant as long as all the gas passes through both apertures; if, however, part of the gas be taken away or absorbed in the space between the apertures, the vacuum will increase in proportion to the amount of gas absorbed. It is evident that, if a manometer or light vacuum gage be connected with this chamber, the amount of gas absorbed will be indicated by the vacuum reading.

The diagram shows the more important parts of the instrument. Gas is drawn from the last pass or uptake of the boiler by means of the

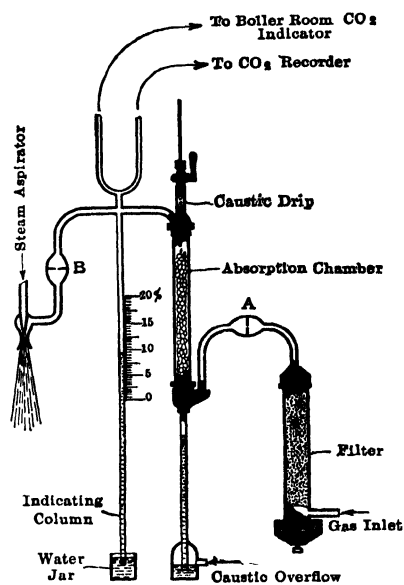


FIG. 251.—Uehling's gas "composimeter."

¹ For the calculation of related quantities see "Heat Balance" (A.S.M.E. Rules), pp. 315 to 322.

aspirator through a preliminary filter located at the boiler, then through a second filter on the instrument as shown, and finally it passes through aperture *A*, the absorption chamber, and aperture *B*, to the aspirator, where it leaves the instrument with the exhaust steam.

Electric Carbon Dioxide Meter.—In essential parts, the electric carbon dioxide meter shown in Fig. 252 consists of two gas cells *A* and *B*, each of which contains a spiral of platinum wire, with provision for supplying electric current to each of the platinum spirals from a storage battery *S*. The cell *A* is filled with air and sealed for a comparison standard. The other cell *B* is arranged so that the sample of flue gas to be tested passes through it. The current from the storage battery *S* is controlled to a constant value by means of the rheostat *R* and the ammeter *A*. The current from the battery *S*, as shown in the figure, passes through the platinum spirals *A* and *B* on one side and the "ratio coils" *C* and *D* on the other. If the gases surrounding the wires are alike in thermal conductivity, heat transmission to the metal walls of the two cells will be equal, and the platinum spirals will be heated to the same temperature.

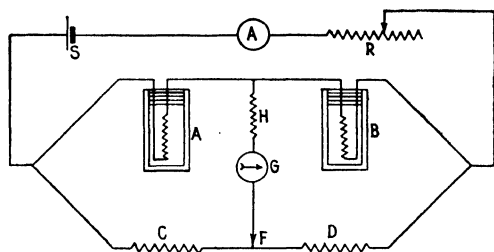


FIG. 252.—Wheatstone-bridge circuit for electric carbon-dioxide measuring device.

The thermal conductivity of carbon dioxide is less than that of air by approximately 40 per cent; and since the other constituents of flue gas, such as carbon monoxide, methane, nitrogen, and oxygen, have substantially the same conductivity as air, the difference in temperature under these conditions between the platinum spirals *A* and *B* will vary with the amount of carbon dioxide in the sample of flue gas being tested.

The arrangement of battery, rheostat, and the four resistances, with the addition of the galvanometer *G* and the resistances *F* and *H*, as shown in the figure, make up a circuit which is suitable for resistance measurements by the Wheatstone-bridge method. In the operation of this device, if the resistances are "balanced" when air is passed through the platinum spiral *B*, the bridge will be thrown out of balance when flue gas containing any appreciable amount of carbon dioxide passes through the platinum spiral *B*, and there results a deflection of the galvanometer *G* nearly proportional to the amount of carbon dioxide in the sample.

Heat is conducted away from the platinum spiral in the cell *B* by the flue gas less rapidly than from the filament in the air cell *A*. With the same flow of electric current in the two platinum spirals, the spiral exposed to the flue gas will be warmer than the one in the air cell. Since the resistance of platinum changes with the temperature, the Wheatstone bridge will be unbalanced as soon as any carbon dioxide is present in the gas cell *B*. The scale on the galvanometer may be marked so that the pointer of the galvanometer will indicate on it directly the percentage of carbon dioxide in the flue gas. This type of apparatus may be made so sensitive that it will indicate a change of $\frac{1}{100}$ per cent of carbon dioxide in air.

An electric *carbon monoxide indicator* has been designed on the same general principles as the electric carbon dioxide meter. The operation of the carbon monoxide meter is based on the catalytic action of a heated platinum wire in producing combustion of carbon monoxide with oxygen. The essential parts of the instrument are two platinum wires which are heated by an electric current. These wires pass through the center of cylindrical chambers, each being in a separate metal block. One of the wires is the "measuring wire," which is in the chamber through which the sample of gas passes, and the other is the "comparison wire," which is maintained at a constant temperature by the electric current. A small amount of air is admitted to the cylindrical chamber surrounding the "measuring wire" to provide oxygen for combustion. The catalytic action causes the carbon monoxide and the hydrogen in the flue gas to burn along the surface of the "measuring wire," thereby raising its temperature above that of the "comparison wire" and increasing the resistance of the "measuring wire" in almost exact proportion to the amount of carbon monoxide and hydrogen gases in the sample of flue gas. The difference in resistance between the "measuring wire" and the "comparison wire" is measured by the Wheatstone-bridge method with a suitable galvanometer. The dial of the galvanometer is calibrated in percentage of carbon monoxide only.

Smoke Determinations.—The method most generally used to determine the density of smoke is with a **Ringelmann chart**, which is shown in reduced scale in Fig. 253. Cards ruled like those shown, but covering a much larger area, are placed in a horizontal row about 50 feet from the observer and in line with the chimney, together with plain white and black cards. The observer glances rapidly from the chimney to the cards and judges which one corresponds most nearly to the color of the smoke. The lines in cards 1 to 4 are, respectively 1, 2.3, 3.7, and 5.5 millimeters thick and the spaces are 9, 7.7, 6.3, and 4.5 millimeters.

A soot-collecting method is sometimes used. It is applied by suspending a narrow plate by a wire from the top of the flue. This plate

is temporarily withdrawn every 2 hours and the collection of soot removed and weighed.

In another apparatus adopted by the Chicago Commerce Association a continuous sample of gas is drawn from the chimney by means of a special Pitot tube and exhaustor. Solid particles in the gas collected are entrapped in a filter. The collecting tube is so arranged that the rate of flow through the apparatus is the same as that through the chimney, so that, when applied to chimneys of different areas, the weight of soot, etc., collected is a measure of the density of the smoke.

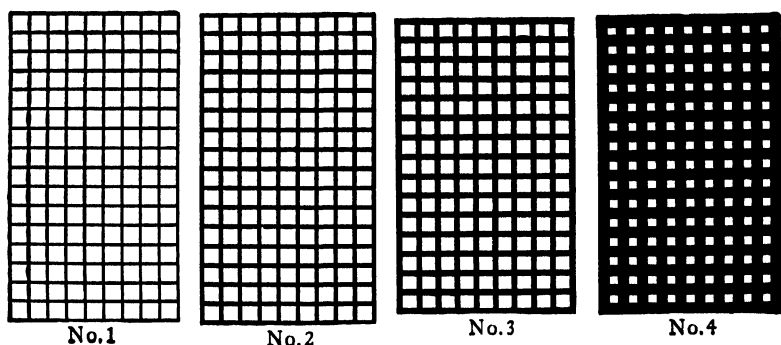


FIG. 253.—Ringelmann smoke chart.

Eddy smoke recorder¹ is one of the few devices for automatically recording the density of smoke. The apparatus consists of a steam ejector which draws a continuous flow of gas from the chimney and discharges it through a nozzle against a paper-covered drum revolved by a clock mechanism. The soot in the smoke makes its own record on the chart.

A.S.M.E. Test Code for Smoke Determination.—In determining smoke density the Ringelmann charts shall be used, although the overall dimensions of the charts used shall be much greater. In taking observations the Ringelmann charts should be placed in a horizontal row, 50 feet from the observer and in a line between him and the chimney. The observer glances rapidly from the chimney to the cards and judges which one corresponds to the color density of the smoke. Readings should be taken every minute, or more often if the variations of smoke density are so rapid as to cause a change of one-half in the smoke-density number on this scale. Careful note should be made of the sky or other background against which the observations are made, as these conditions affect the apparent color of the smoke.

Readings should be recorded according to the scale of values shown in the table on page 287.

It is to be noted that the foregoing rules apply to kinds of smoke in which the particles are black. In some cases, however, the smoke may be inherently of some other color, as, for instance, some shade of brown. For such cases the same charts and method shall be used. The observer will now grade the smoke by number accord-

¹ Hamler-Eddy Co., Chicago.

ing to its density in terms of the base color other than black, whatever it is. For instance, if the color be brown, a smoke of 100 per cent brown smoke shall be recorded as card No. 5, etc.

Card	Smoke density, per cent	Description
No. 5 (black).....	100	Dense black
No. 4.....	80	Medium black
No. 3.....	60	Dense gray
No. 2.....	40	Medium gray
No. 1.....	20	Light gray
Between No. 0 and 1.....	Trace	Trace
No. 0 (white).....	0	Clear

Photoelectric Cell as Smoke Detector.—Methods have been developed for the amplification of photoelectric impulses by means of radio vacuum tubes. The photoelectric impulses can be amplified as much as a million times by means of a single vacuum tube, having in view a power output sufficient to operate directly the usual type of mechanical relay. A device of this kind, as shown in Fig. 254, is a combination of what is practically a standard three-electrode vacuum tube with a photosensitive control electrode. The lower part, resembling an ordinary radio vacuum tube, consists of an oxide-coated filament which operates at a temperature below visual emission and the usual grid and plate, all mounted on a standard base. One-half of the inside of the upper part of the cell is coated with a substance, such as an alkali metal, which emits electrons under the influence of light. To prevent any possibility of light from the filament falling on the sensitive coating, the upper part of the cell is shielded from the amplifying portion of the tube by metal shields, which serve also to connect the grid to the coated surface of the upper part of the cell. The electron collector, shown as a loop in the upper part of Fig. 255 is connected to the plate of the amplifying element. Great care is taken to prevent the alkali metal coating from condensing appreciably on the insulating parts of the cell. The connections for this cell when used as a smoke detector are shown in Fig. 255. When in the dark, the action of the upper part of the cell is negligible, and the lower part acts as an ordinary vacuum tube. The very high negative charge (grid bias) impressed on the grid by the "C" battery completely neutralizes the attraction of the positive charge on the more distant plate for the

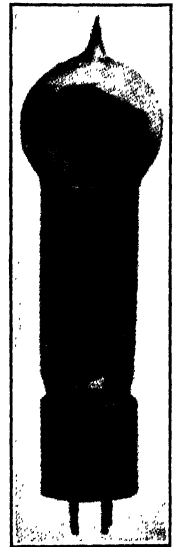


FIG. 254.—
Radio vacuum
tube for smoke
detecting device.

electrons given off by the filament, so that the current in the "plate" or "B" battery circuit is practically zero.¹ As long as no current is flowing in the grid circuit, the effect of the high resistance in this circuit is also negligible.

If, however, light falls upon the sensitive coating, electrons are given off and these are attracted to the collector by its positive charge, which equals the sum of the B and C battery potentials. The resultant current is proportional to the brightness of the light which falls on the sensitive coating, being independent of the applied voltage, provided this is sufficiently great to attract to the collector all the electrons that are given off. This current in the light sensitive part of the cell is very small, being only a fraction of a microampere. It does not have any appreciable effect on the relay. It is, however, sufficient to produce a voltage drop over

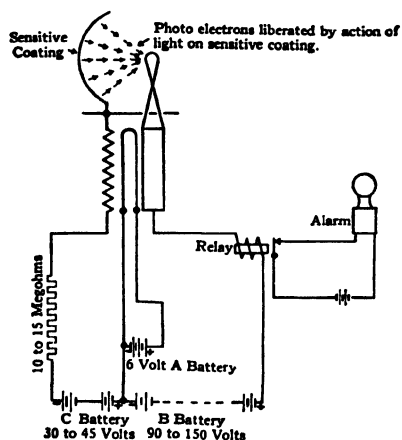


FIG. 255.—Diagram of photoelectric device for smoke observations.

the high resistance in the grid circuit and partly or even completely to neutralize the negative grid bias on the three-electrode amplifier, permitting a flow of current in the amplifier plate circuit which is many times that in the upper part of the cell. This current is still quite small (several milliamperes) but is sufficient to actuate the relay. Within limits, this amplifier plate current is proportional to the amount of light which falls on the light sensitive element of the upper cell. There is, of course, a certain time lag between the fall of light on the sensitive coating and the resultant flow of current in the plate circuit of the amplifier.

This time lag is less than one ten-thousandth of a second.

When used as a smoke detector, this cell is located in the end of a tube to prevent light from an outside source affecting it. A beam of light is passed over the apparatus to be protected and falls on the sensitive coating, thus maintaining current in the relay circuit and hence keeping the alarm circuit open. The alarm circuit, such as a bell, may be of any desired type which can be operated from a relay. Any smoke arising from the apparatus will diminish the intensity of the beam of light, causing the relay to actuate the alarm. The types of apparatus that depend for their results on color are not correct indicators of the amount of white or nearly white solids in the samples tested, as, for example, nearly white ashes.

¹ For further information regarding radio vacuum tubes and photoelectric cells, see Moyer and Wostrel, "Practical Radio," 4th ed., pp. 51 to 77 and 365 to 378.

CHAPTER X

A.S.M.E. CODE ON GENERAL INSTRUCTIONS REGARDING POWER PLANT TESTS

Objects.—Ascertain the object of the test. This object should be kept in view during preparations for the test and in the conduct of the test itself. The methods to be used and the accuracy sought should be in accord with the object in view.

A.S.M.E. Power Test Codes.—The Power Test Codes as given here and in the other chapters have been adopted by the A.S.M.E.; nevertheless all codes are potentially subject to revision to adjust them to changing conditions of power plant operation. The codes for Steam Boilers and other Steam Generating Units, Steam Engines, Steam Turbines, Internal Combustion Engines, Displacement Compressors and Blowers, Refrigerating Plants, Condensers, Feed-water Heaters, Centrifugal and Rotary Pumps have been completed and it is unlikely that there will be important changes. In some of the codes the parts relating to tests which are not often made are abridged.

Among the many objects of performance tests, the following may be noted: (a) determination of capacity, efficiency, or regulation, and the comparison of these with standard or guaranteed results; (b) comparison of different conditions or methods of operation; (c) analysis and interpretation of plant performance; (d) comparison of different kinds of fuel; (e) determination of the effects of changes of design or proportion upon capacity or efficiency.

If questions of fulfilment of contract are involved, there should be a clear understanding in writing between all parties as to the operating conditions which should obtain during the trial, the methods of testing to be followed, corrections to be made in case the conditions actually existing during the test differ from those specified, and all other matters about which dispute may arise, unless these are already expressed in the contract itself.

Preparations.—Ascertain and record manufacturer's serial number, plant designation, or other means of identification of each unit involved in the test.

Examine and record the general features, arrangement, and condition of the apparatus and plant, and, if needed, make sketches to show the arrangement and any unusual features. As far as possible, examination should be internal as well as external.

Determine the principal dimensions of the apparatus to be tested, particular attention being given those bearing on the objects in view.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects of operation tending to make the result unfavorable should first be remedied.

In all tests in which the quantity of gas, vapor, or liquid flowing in pipes or ducts is determined, or in which such quantity forms an item in

the calculation of results, precautions must be taken to guard against leakage. Such precautions are necessary in the case of blow-off connections, drips and drains, cross-connections, isolating valves, packing and joints of all kinds, etc. All outlets and inlets other than those actually required in the test should be *blanked off* or so arranged that any leakage in or out can be observed. A leakage test to determine tightness and to measure unpreventable leaks should be made in every instance.

Instruments and Testing Apparatus.—The choice of instruments and apparatus for any test must be determined by local conditions, adaptability, and availability, but in all cases the types called for in the various codes should be used. In making this choice, the degree of accuracy desired should be one of the guiding factors.

It should be noted that the readings of various instruments are generally combined to obtain a given result, and that under such conditions the effect of the probable error of each instrument upon this result should be considered in making a choice. It should also be noted that the measurement in any test which is least capable of accurate determination often determines the accuracy obtainable in the statement of the results of the test, and that under such conditions the attempt to measure other values to greater degrees of accuracy may be simply a *waste of time and effort*.

All instruments and testing apparatus used should have designating numbers or other means of identification firmly attached and such identifications should be entered on the log sheets. It is advisable to mark on this identification tag a statement indicating the degree of accuracy to which the instrument is to be read, for example, "Read to the nearest 2 degrees." This will assist in eliminating needlessly exact as well as seriously inexact data.

All instruments and testing apparatus should be calibrated, and any which may give trouble if not approximately correct and all which may be broken in use should be calibrated in advance. Additional calibrated instruments should be provided to replace any which may be broken during the test. Any instrument which may not maintain its calibration throughout a test should be calibrated before and after the test and, if necessary, at intervals during its progress. Instruments and testing apparatus should always be so arranged and installed as to give maximum accessibility and best possible illumination in order to facilitate use.

Notebooks or log sheets should be so arranged that all readings of one group of instruments or all readings made by one observer appear in one book or on one group of sheets. The pages or sheets upon which entries are to be made should be ruled in advance, should be given proper identification marks, should contain space for the date and the name of the observer, and should contain proper headings and subheadings. All such sheets should have at least one column in which time is recorded.

When rates are to be determined by differences between observations, a time column should accompany the observation column. Space for remarks should be provided on every sheet.

A desk or table from which the tests may be directed and at which the records may be assembled should be provided in tests requiring many observers. This should be located with due regard to the positions they occupy. A signal system or a system of intercommunication is convenient and may be provided.

The person in charge of a test should have the aid of a sufficient number of assistants to permit him to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications, and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs to vitiate the data. In tests extending over long periods of time, there should be a captain or supervisor for each shift, this supervisor to act as the person in charge as just outlined.

Starting Conditions.—It is advisable to make one or more preliminary tests to determine the adequacy of the instruments and apparatus and of the training of the personnel. When conditions do not permit such preparatory runs, operations may be started and the time at which conditions become satisfactory can be chosen later as the starting time of the test. Before the test begins or before continuing the test after an important change of conditions during operation, the apparatus should be run under normal conditions for a sufficient length of time to bring about equilibrium with respect to thermal or other factors, except in cases in which the effects of the variation of such conditions are to be determined or included. These exceptions include such effects as result from starting and stopping, holding units ready for start, and banking fires.

In preparation for a test to demonstrate maximum efficiency, it is desirable to make preliminary tests to determine the most advantageous conditions.

Operating Conditions.—In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity should be made unless this uniformity corresponds to the regular practice, and when this is the object the usual working conditions should prevail throughout the trial.

Records.—Records of a test are divided into two classes: (a) records of preliminary examination and calibration; (b) records of observations made, and events occurring during the test.

Records of both classes should be made with an original and a sufficient number of carbon copies so that one copy may be given to each interested party. All records should be signed by the respective observers and countersigned by the person in charge of the test.

A log of the observations made during the test should be entered in the notebooks or on the sheets in such manner that the test may be divided into suitable periods to show the degree of uniformity attained.

The readings of instruments from which averages are to be obtained should be recorded at intervals fixed by the extent of their variations. Intervals of from 15 to 30 minutes are usually sufficient, but there are cases in which intervals of less than a minute are necessary on account of rapid and irregular variations.

Ordinarily the intervals between readings should be regular, but in some cases such regularity is not required and possibly not even desired.

It is not always necessary to read all instruments at the same intervals, nor is it always necessary to read simultaneously those which are read at the same intervals. In cases in which the average of a long series of readings is used in calculating results, regularity rather than the attainment of simultaneous readings should be the guiding principle. In cases in which each set of observations is used for calculating a result and the results are then averaged, simultaneous readings are all-important. When observations are made to determine rates by sums or differences, the exact time of making the observation is necessary. Whenever possible, check readings should be made after an observation has been recorded. In case of error in a recorded observation no erasure should be made. A line should be drawn through the incorrect entry, the correct reading should be recorded above, and an explanation entered under remarks.

The records made during the test should show the extent of fluctuation of the various instruments in order that data may be available for determining the effect of such fluctuations on the accuracy of the calculated results. A record should never consist of a series of tallies as is sometimes made when loads of given weight are discharged at odd intervals or under similar repeated conditions. In such cases each weight and the time when taken should be recorded.

Observations of basic character in a given test should be made and recorded in duplicate by two observers when the importance of the test warrants such precautions. In order that the readings made by these observers may be in accord before it is too late, the readings should be compared and an agreement reached soon after the observations are made.

A running **graphical log** of the principal quantities recorded during a test should be maintained. Such graphical records serve to call attention to undesirable or unintentional lack of uniformity in the conduct of operations and to detect errors of observation or record before it is too late to correct them. Such graphical logs should be arranged so that time is plotted horizontally and quantities vertically as in Fig. 256.

Profile paper will be found very convenient for the plotting of graphical logs, and where rates are concerned it is advisable to use "Aritholog" paper. The latter is divided so that the vertical scale is logarithmic, while the horizontal scale is arithmetic.

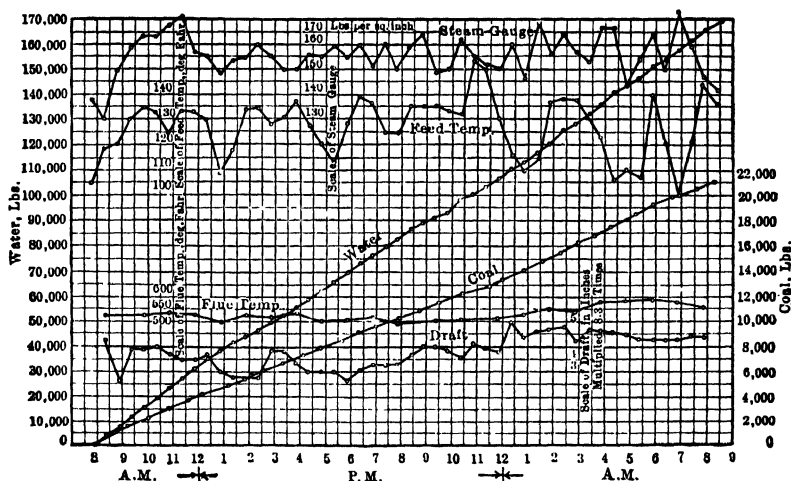


FIG. 256.—Graphical chart of a boiler trial.

Every event connected with the progress of a test, however unimportant it may appear at the time, should be recorded on the proper log sheets with the time of occurrence and the name of the observer who noted the event and entered the record.

Working up the Data.—It is desirable to work up the data *approximately* during the progress of the test. This tends to call attention to omissions and to irregularities of various parts. Calculations of this kind are, however, to be regarded as auxiliary testing methods and the results obtained should not be used in the preparation of the final report unless the test is intended to be of approximate nature and the result to be so given. All parts of the record which show evident irregularities must be either corrected or eliminated in order that the final results may not be knowingly in error; and if satisfactory adjustments cannot be made, the test must be repeated. All adjustments of this type should be noted and explained in the final report.

Two distinctly different methods are available and it is necessary to recognize the conditions under which each may be used. These

methods are: (a) the method using primary averages, in which all readings of each instrument are averaged for a given period and these averages are then used in computing final results; and (b) the method using final averages, in which the final result is calculated from each set of simultaneous observations and the final results are then averaged as a grand average.

The choice between these two methods of computing results should be determined by the type of formula involved and the degree of accuracy desired. When the formula involves the sum or difference of first powers of the observed quantities, either method will give the same numerical results. When the formula involves the product or the quotient of first powers, or fractional powers or powers greater than the first, the method of primary averages will generally give approximate results only.

In working out results and in making¹ reports, the principles of the theory of errors may be applied to determine the observations which should be rejected and the manner of treating the data obtained, and to assign the relative importance of the various quantities determined.

Degree of accuracy attainable is of great significance. Attainable accuracy of results is determined primarily by the accuracy of the observations upon which the results rest. There are three distinctly different sources of error to be considered exclusive of gross blunders. These are: (a) instrumental; (b) errors of observation conforming to the laws of probability; and (c) errors resulting from the personal equation of the individual observers.

The instruments used in testing always have definite limits of accuracy. Calibration does not affect this property. The correction factors determined by calibration merely serve to correct the indications of the instrument by such an amount as to bring the results to the degree of accuracy of which the instrument is capable.

For example, a given temperature-measuring device may be so constructed that it can be read to the nearest 0.1°F. Calibration of this thermometer may show that it reads 0.5°F. high at one part of the scale and that the error gradually changes along the scale until it reads 0.2°F. low at another distant point. These correction factors are accurate only to the first decimal point, as given, and the fact that the thermometer has been calibrated does not make it possible to read closer than 0.1°F. Calibration does, however, convert this device into an instrument which can be used for reading temperatures to the nearest 0.1°F., which might be called the scale of accuracy of the instrument. Again, although this thermometer may have the properties indicated above, it may be used under such conditions that it cannot respond to temperature changes as rapidly as these occur in the medium, the temperature of which is being measured. Obviously, this would introduce another type of instrumental error. Moreover, this error is of such a type that the scale accuracy of the instrument is

¹ For a discussion of the theory of errors, see Marks, "Mechanical Engineers' Handbook," 2d ed., pp. 119-121.

really not usable, unless by chance temperature changes happen to occur in such a way that the plus and minus errors cancel.

The property discussed might be referred to as the *sensitivity* of the instrument with respect to time. Further, the instrument might be so constructed or in such a condition that, once having attained a position of rest, it might require more than a 0.1°F. temperature change to cause it to move to a new temperature indication. This property may be referred to as the sensitivity of the instrument with respect to displacement or motive impulse. Obviously, the scale accuracy of the instrument may not be attainable if the instrument is not sufficiently sensitive in both respects referred to above. It is not, therefore, sufficient to consider the scale accuracy of instruments in an analysis of the accuracy attainable. It is necessary to consider the accuracy attainable under the given conditions of use.

Errors of observation conforming to the laws of probability are necessarily both positive and negative and with a sufficient number of observations of a given quantity they will practically cancel. Under the conditions prescribed for the Power Test Codes the time intervals between readings and the length of tests are intended to be such that errors of this kind will cancel to the extent required.

Errors due to the personal equation of the observer are of many sorts and in a sense are not subject to correction. Some observers are noticeably quicker than others and will therefore yield more accurate results when dealing with quantities in a state of flux. Some observers consistently read a given instrument low and others consistently read it high. Some observers can estimate readings between scale divisions more accurately than others. Some observers have so persistent a memory for past observations that they unconsciously allow this memory to influence subsequent observations, while others seem to be absolutely unaffected in this respect. No method can be prescribed for guarding against errors arising from this source. The best that can be done is to distribute the available observers to the best advantage and to check the performance of each before the test or periodically during the test by having several observers make the same determinations independently.

These various sources of error should be studied before a test is started and they should be given proper consideration during the conduct of the test. If such precautions are taken, the probable accuracy of the different groups of observations will be known when these are used in working up results. It is then necessary to determine the way in which the known degree of accuracy will affect the results of the different calculations which have to be made. **Accuracy of observations** is commonly referred to **percentage**—for instance, a given observation is said to be accurate within plus or minus 1 per cent. It is more convenient to inter-

pret this form of expression in terms of significant figures, as it makes the attainable accuracy more obvious.

In general, that numerical value having the minimum number of **significant figures** determines the maximum number of significant figures which need be retained in any of the numerical values entering into a calculation and also determines the maximum number of significant figures which should be included in the results. In dropping significant figures to adjust to the accuracy of that numerical value having the minimum number, it is proper to retain a number of figures, one greater than this minimum, until the final result is obtained. This is done to prevent a possible cumulative error. In dropping significant figures in making adjustments of this sort, one unit is added to the last figure retained if the first discarded figure is 5 or more.

The effect of using the general rule for significant figures given in the preceding paragraph should be studied in each case to make certain that it is completely applicable.

In recording results of tests it is useless and often misleading to give figures which indicate a greater degree of accuracy than that actually attained. Results should be recorded in such a way as to indicate clearly their **probable accuracy**. Thus the statement that the water rate of a certain turbine and generator under certain steam conditions, a certain back pressure, and a certain load is 12.1 pounds per kilowatt-hour is incomplete. It would be complete if expressed as 12.1 plus or minus 0.1 or 12.1 plus or minus 0.05 pounds or other combination which indicates the limits of accuracy.

In recording results in this way it is particularly necessary to maintain a complete view of the situation and not to be misled by the accuracy attainable with the instruments and methods of test used in a given case. For instance, it might be possible to conduct a complete boiler test in such a way that the figure attained for overall efficiency is accurate to ± 1.0 per cent, in so far as the indications of instruments and the handling of observations are concerned; but tests with different firemen or with the same firemen on different days might show efficiencies varying by as much as ± 2.0 per cent or more for the same rate of steaming and with all other conditions as nearly the same as possible. Obviously, such tests should be reported in such a way as to indicate their true worth. This might be done by stating in the report that in so far as instrumental and other similar factors are concerned the result is accurate to within ± 1.0 per cent and that it is believed to be representative of the best attainable efficiency within $+1.0$ and -1.0 per cent. Or the report might state that, with the equipment in the condition under which it was tested and with the personnel operating that equipment during the test, a result representing the true value to ± 1.0 per cent was obtained. It is essential to bear in mind the fact that to most persons numbers mean exactly and completely what they read and that this fact has particular significance in litigation.

It is also essential to note that the sort of averaging which is done by drawing a line or a curve which seems best to average the results of a test is good or bad, depending upon the kind of results which are averaged. For instance, if a boiler is tested at, say, five different rates of steaming, and only one test is made at each rate, the drawing of a so-called "aver-

age" curve or line means little. On the other hand, if three tests are made at each rate of steaming, a curve drawn so as to take account of the average of each group of plotted points is of real significance.

Report.—The report of a test of considerable magnitude should be divided into three distinct parts, as follows: (1) brief statement of object, results, conclusions, and recommendations; (2) complete presentation of the leading facts regarding the entire work; (3) appendices giving details such as methods of calculation, methods of calibration, descriptions of special testing apparatus, results of preliminary or special tests, etc.

The complete statement referred to in Item 1 above should be subdivided as follows: (a) object, giving authority, preliminary agreements, and other relevant information; (b) description of apparatus tested; (c) methods of testing, giving conditions of operation, location, arrangement and method of using instruments and testing apparatus, and personnel; (d) discussion of the data and results both as to accuracy and as bearing on the object of the test; (e) leading conclusions; (f) tables of data and calculated results or graphical expositions of the same; (g) charts not included under Item (f).

Reports of tests of lesser magnitude should follow the same general plan.

In preparing a report the work should be so done that the report will be self-contained in all respects. The best test of such a requirement is to assume that the report may be brought into court in connection with litigation involving the object for which the test was made. If the report can stand alone under such conditions, it may be considered complete.

The standard units on which to base the various measures of capacity and the standard forms of expressing efficiency and economy to which the codes apply are as follows, as defined in the A.S.M.E. codes:

Apparatus	UNITS OF OUTPUT	Units
Stationary boilers and other steam-generating units.....	(a)	Heat absorbed by water and steam, in B.t.u. per hr. ¹
	(b)	Actual evaporation, in lb. of steam per hr., at observed steam pressure and quality, or temperature, and observed feed-water temperature
Reciprocating steam engines.....	(a)	I.hp. at observed conditions of steam supply and exhaust
	(b)	B.hp. at observed conditions of steam supply and exhaust
Steam-engine generators.....		Net kw. at generator terminals at observed conditions of steam supply and exhaust
Steam turbines.....		B.hp. at observed conditions of steam supply and exhaust
Turbine generators.....		Net kw. at observed conditions of steam supply and exhaust
Pumping machinery.....	(a)	Cubic feet discharged per second at observed total suction and discharge pressures
	(b)	Gallons discharged in 24 hr. at observed total suction and discharge pressures
	(c)	Gallons per minute at observed total suction and discharge pressures
	(d)	Water-horsepower output at observed total suction and discharge pressures
Compressors and blowers, centrifugal and displacement.....	(a)	Cu. ft. of free air (or other gas) per min., at observed total intake pressure and temperature delivered at observed total discharge pressure
	(b)	Air-hp. output at observed inlet and delivery conditions (p. 441)
Gas producers.....	(a)	Lb. of fuel as fired per hr., of observed high calorific value (pp. 242 and 311)
	(b)	Hot-gas output in cu. ft. of dry gas per hr., at observed temperature and pressure, and observed high calorific value
	(c)	Cold-gas output in cu. ft. of dry gas per hr. at 68°F. and one standard atmosphere
	(d)	Heat output per hr. in hot gas
	(e)	Heat output per hr. in cold gas
Gas and oil engines.....	(a)	B.hp.; (b) I.hp. ²
Hydraulic turbines.....		B.hp.
Condensers.....		Heat transferred per hr., at observed vacuum, inlet and outlet circulating-water temperatures, and cu. ft. of gases discharged by air pump per hr., measured at 68°F. and 1 standard atmosphere (p. 444)
Evaporators.....	(a)	Heat transferred per hr. at observed temperature of evaporation, and observed temperature and pressure of steam supply

Apparatus	Units
	(b) Lb. of raw liquor per hr.
	(c) Lb. evaporated per hr.
	(d) Lb. of concentrate per hr.
Feed-water heaters and fuel-oil heaters.....	Heat transferred per hr. at observed steam pressure and temperature and at observed inlet and outlet water or oil temperatures ¹
Economizers.....	Heat transferred per hr. at observed pounds of flue gases per hr., inlet gas temperature and inlet and outlet water temperatures ²
Cooling towers and cooling ponds	Heat dissipated per hr. at observed inlet and outlet water temperatures, observed air temperatures and humidity
Refrigerating machines.....	(a) Heat absorbed per hr. at observed head pressure and observed suction or cooler pressure ³
	(b) Commercial tons of refrigeration at observed head pressure, and observed suction or cooler pressure

¹ The rate of heat absorption of a stationary steam-generating unit is its output in B.t.u. per hour. For very large units, it is sometimes convenient to use multiples of B.t.u. as the unit. Such multiple units are $kB = 1,000$ B.t.u. and mB equals 1,000,000 B.t.u.

Boiler horsepower was originally standardized by the A.S.M.E., as explained on p. 304. At present, however, steam rates vary from 8 pounds in large condensing turbines to 50 or 60 pounds per horsepower-hour for small noncondensing units, so that the term "boiler horsepower" has no longer any real connection with the steam consumption of the engine. The term "boiler horsepower" has never been used in England or in Continental countries.

The rating of steam boilers for size only in terms of square feet of heating surface is still recommended by the A.S.M.E.

Heat absorbed by water and steam $Q = w_s(h_1 - h_f)$ B.t.u. per hour, where w_s = total pounds of water evaporated per hour; h_1 = heat content (enthalpy) of the steam generated, in B.t.u. per pound; and h_f = heat content (enthalpy) of feed water, in B.t.u. per pound.

A subsidiary unit which may be used for stationary boilers is a "boiler horsepower," or $34\frac{1}{2}$ pounds of water evaporated from and at 212°F. per hour, that is, from water at 212°F. into steam at the same temperature. Electrical engineers have suggested a unit termed "myriawatt," which differs but little from boiler horsepower when expressed in B.t.u. per hour.

² Net indicated horsepower for internal-combustion engines is the indicated horsepower of the main cylinders minus the indicated horsepower of the auxiliary cylinders for scavenging and injection. Whenever possible, the net output should be in terms of brake horsepower, thus to eliminate many correction factors.

Indicated horsepower of an internal-combustion engine is not a true output, as it does not represent the power available for use.

³ In apparatus such as surface condensers and evaporators, both sides of the apparatus contain fluids, and measurements may be made on either of the fluids. If the apparatus is well insulated or is at nearly the same temperature as its surroundings, so that the external heat transfer is negligible, the rate of heat transfer will be the same when determined in either of the fluids, but different methods may be suitable in different cases.

UNITS OF PERFORMANCE

Apparatus	Quantities and Units	Conversion Relations
Stationary boilers and other steam-generating units	(a) Efficiency of stationary steam-generating unit: ratio of heat units output to high calorific value of fuel as fired	
Solid fuel only	(b) Rate of combustion in B.t.u. per sq. ft. of grate surface per hr. as fired ¹	
All fuels	(c) Rate of combustion, in lb. of fuel as fired per hr. per cu. ft. of furnace volume ²	
	(d) Heat transferred per sq. ft. of heating surface per hr. (see footnote ³ , p. 299)	
	(e) Heat developed per cu. ft. of furnace volume per hr. ² (p. 303)	
Reciprocating engines . . .	(a) Heat rate, in B.t.u. per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(b) Thermal efficiency referred to (1) 2,545 ÷ Item a i.hp.; (2) b.hp.	
	(c) Engine efficiency (Rankine) referred to (1) i.hp.; (2) b.hp.	
	(d) Steam rate, lb. of steam per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(e) Mechanical efficiency	
Steam turbines	(a) Heat rate, in B.t.u. per b.hp.-hr.	
	(b) Thermal efficiency	2,545 ÷ Item a
	(c) Engine efficiency (Rankine)	
	(d) Steam rate, in lb. of steam per b.hp.-hr.	
Steam-engine generators and turbine generators	(a) Heat rate, in B.t.u. per net kw.-hr.	
	(b) Thermal efficiency	3,413 ÷ Item a
	(c) Engine efficiency (Rankine) referred to net kw.	
	(d) Steam rate, in lb. of steam per net kw.-hr.	
Steam-driven pumping engines	(a) Heat supplied per water hp.-hr.	
	(b) Thermal efficiency referred to water hp.	2,545 ÷ Item a
	(c) Steam rate, in lb. of steam per water hp.-hr.	
	(d) Mechanical efficiency (reciprocating pumps only)	
Steam-driven compressors, blowers and fans	(a) Heat rate, in B.t.u. per air hp.-hr.	
	(b) Thermal efficiency	2,545 ÷ Item a
	(c) Steam rate, in lb. of steam per air hp.-hr.	
	(d) Steam rate per unit of volume	
	(e) Gross	} hp. per cu. ft. per min., at intake conditions
	(f) Net	
	(g) Indicated	

Apparatus	Quantities and Units	Conversion Relations
	(h) Volumetric efficiency, for displacement compressors only	
	(i) Compression efficiency, for displacement compressors only	
	(j) Mechanical efficiency, for displacement compressors only	
	(k) Overall efficiency	
Direct-drive steam plants	(a) Fuel rate, in lb. of fuel as fired per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(b) Steam rate, in lb. of steam generated per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(c) Heat rate, in B.t.u. per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(d) Thermal efficiency referred to (1) i.hp.; (2) b.hp.	$2,545 \div \text{Item } c$
Steam-electric plants	(a) Fuel rate, in lb. of fuel as fired per net kw.-hr.	
	(b) Steam rate, in lb. of steam generated per net kw.-hr.	
	(c) Heat rate, in B.t.u. per net kw.-hr.	
	(d) Thermal efficiency, overall, referred to net kw.	$3,413 \div \text{Item } c$
Steam pumping plants . . .	(a) Fuel rate, in lb. of fuel as fired, per water hp.-hr.	
	(b) Steam rate, in lb. of steam generated per water hp.-hr.	
	(c) Heat rate, in B.t.u. per water hp.-hr.	
	(d) Thermal efficiency referred to water hp.	$2,545 \div \text{Item } c$
Steam air-machinery . . .	(a) Fuel rate, in lb. of fuel per air hp.-hr.	
	(b) Steam rate, in lb. of steam generated per air hp.-hr.	
	(c) Heat rate, in B.t.u. per air hp.-hr.	
	(d) Thermal efficiency referred to air hp.	$2,545 \div \text{Item } c$
Gas producers	(a) Hot-gas efficiency; ratio of high calorific value plus sensible heat above room temperature in hot gas to the high calorific value of fuel as fired	
	(b) Cold-gas efficiency; ratio of high calorific value of gas to high calorific value of fuel as fired	
Internal-combustion engines	(a) Fuel rate, in lb. of fuel as burned per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(b) Fuel rate, in cu. ft. of gas of stated high calorific value at 68°F. and one standard atmosphere, (p. 241), per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(c) Heat rate, in B.t.u. per (1) i.hp.-hr.; (2) b.hp.-hr.	
	(d) Thermal efficiency referred to (1) i.hp.; (2) b.hp.	$2,545 \div \text{Item } c$

Apparatus	Quantities and Units	Conversion Relations
Gas- or oil-electric units . .	(a) Fuel rate, in lb. of fuel as burned per net kw.-hr. (b) Fuel rate, in cu. ft. of gas of stated high calorific value at 68°F. and one standard atmosphere per net kw.-hr. (c) Heat rate, in B.t.u. per net kw.-hr. (d) Thermal efficiency	3,413 ÷ Item c
Gas-producer plants	(a) Fuel rate, in lb. of fuel <i>as fired</i> per net (1) i.hp.; (2) b.hp. (see footnote, p. 315) (b) Heat rate, in B.t.u. per net (1) i.hp.-hr.; (2) b.hp.-hr. (c) Thermal efficiency referred to (1) i.hp.-hr.; (2) b.hp.-hr.	2,545 ÷ Item c
Hydraulic turbines	Efficiency of turbine; ratio of b.hp. to water hp.	
Hydraulic turbine generators	Efficiency of unit; ratio of net electrical hp. to water hp.	
Surface condensers	(a) Heat transferred per sq. ft. of cooling surface under observed conditions of vacuum, inlet and outlet circulating-water temperatures and cu. ft. of gases discharged by air pump per hr., at 68°F. and 1 standard atmosphere (p. 444) (b) Heat-transmission coefficient	
Evaporators	(a) Heat transferred per hr., with observed conditions of steam supply, temperature and pressure (b) Lb. of steam per lb. of distillate (c) Lb. of steam per lb. of concentrate (d) Heat-transmission coefficient	
Closed feed-water heaters and fuel-oil heaters	(a) Heat transferred per sq. ft. of heating surface per hr. at observed conditions of steam supply, and stated inlet and outlet temperatures of water or oil (b) Heat-transmission coefficient	
Oil coolers	(a) Heat transferred per sq. ft. of cooling surface per hr., at observed inlet and outlet oil temperatures, and observed inlet and outlet cooling-water temperatures. (b) Heat-transmission coefficient	
Economizers	(a) Heat transferred per sq. ft. of heating surface per hr. at observed lb. of flue gases per hr., inlet flue-gas temperature and inlet and outlet water temperatures (b) Heat-transmission coefficient	

Apparatus	Quantities and Units	Conversion Relations
Steam-driven refrigerating machines	(a) Heat supplied in steam per ton of refrigeration, at observed conditions	
	(b) Coefficient of performance; ratio of heat abstracted to indicated work, in B.t.u.	
	(c) Steam rate, in lb. of steam per hr. per standard ton of refrigeration, at observed conditions	

¹ Grate surface is the horizontal area within the four walls surrounding the fuel-supporting structure.

² The total furnace volume for horizontal return-tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between the tubes. It includes the volume behind the bridge wall as in ordinary horizontal return-tubular boiler settings, unless this space behind the bridge wall is obviously ineffective as in the case of waste-heat boilers with auxiliary coal furnaces.

CHAPTER XI

TESTING OF BOILERS AND OTHER STEAM-GENERATING UNITS

Leakages of any kind are always a lurking enemy for those engaged in any kind of accurate testing, and work with boilers is no exception. Poor results with boilers are due more often to air leakage than to any other fault. Air entering the setting and flues instead of the furnace does not assist combustion but, on the contrary, absorbs from the hot gases a quantity of heat which otherwise might pass through the boiler-heating surfaces into the water in the boiler. When the object of a series of tests is, for example, to compare one kind of coal with another, or one type of grate or mechanical stoker with another, the losses due to air leakage would not be of much consequence in what are only comparative results; but if a boiler is to give the best possible efficiency and capacity, air leaks must be stopped.

In practice the importance of closing air leaks in the boiler setting is forcefully presented when patented devices for fuel saving are installed in boiler plants. Important economies in many cases are guaranteed if the new device is adopted, and then the claims of the agent are made good by instructing his workmen to go over the boiler, closing up all cracks in the setting through which cold air could enter, and at the same time covering the outside surface of the setting with a coating impervious to air. By such means the owner of the plant pays a high price for results that could have been obtained much more cheaply.

In steam-boiler practice "boiler" horsepower is still a method of rating. The term "horsepower" has two very distinct meanings in engineering practice. Usually it is taken to mean the rate of doing work or the work done in a definite period of time. In this sense it means, as in the case of engines, turbines, water wheels, etc., 33,000 foot-pounds per minute. In a steam boiler, however, where the work done must be measured by the conversion of water into steam, a horsepower was formerly taken as the evaporation of 30 pounds of water at 100°F. into steam at 70 pounds pressure above the atmosphere. When this unit was adopted, it was considered that 30 pounds per hour was approximately the requirement per indicated horsepower of an average engine. The Committee on Boiler Tests of the A.S.M.E. adopted at one time what is in effect the same unit, stating it, however, somewhat differently—that a **boiler horsepower is equivalent to evaporating 34.5 pounds of steam¹ per**

¹ When the steam supply is small or if, for any reason, the noise due to escaping steam from the calorimeters used is objectionable, a calorimeter may be shut off some-

hour from feed-water temperature of 212°F. into steam at the same temperature. According to the steam tables (page 593) this is equivalent to approximately 33,480 B.t.u. per hour, or 558 B.t.u. per minute.

Unit of Evaporation.—For reducing the results of the boiler tests to a common standard, the term “unit of evaporation” was formerly used. It is the heat required for the “absorption of 1,000 B.t.u. per hour.” It has been variously defined and was once the heat required “to evaporate a pound of water from and at 212°F.” (approximately 970.2 B.t.u.).

Graphical log sheets of boiler tests similar to the one shown in Fig. 256 are very serviceable for checking the observations when made during the test as the data are taken. In the report of a test they show also the relative irregularity or regularity of the conditions affecting the results.

Standard Methods for Boiler Trials.—The A.S.M.E. has adopted rules for conducting boiler trials which are generally accepted in America and are also considered with favor in England.¹

A.S.M.E. TEST CODE (1929) FOR BOILERS AND OTHER STATIONARY STEAM-GENERATING UNITS

INCLUDING STOKERS, SUPERHEATERS, ECONOMIZERS, REHEATERS, AND AIR HEATERS

The Test Code for Stationary Steam Boilers applies to the boiler, water walls, water floor, water screen, fuel-burning apparatus of whatsoever nature, such as superheater, reheater, air heater, and economizers, but not to apparatus required for their operation, such as engines or motors for driving stokers, feed pumps, draft fans.

Economizers² are considered as part of the generating units. They may be divided into two general classes: (a) *separate economizers*, being those with their own housing and connected to the boilers with flues, and (b) *integral economizers*, which are contained within the boiler setting and through which a part of the boiler circulating water may or may not pass.

Observations made when steam boilers equipped with economizers of the first-named class are being tested, may readily be separated so as either to include or to exclude the economizer in the results. In the second class, however, it is practically impossible to segregate the economizer from the boiler, and tests of units so equipped should be conducted and reported as if the economizer is a part of the boiler proper.

times between readings. The length of time the calorimeter is in operation must then be carefully noted in order to determine the weight of steam lost through it by calculating the flow through its orifice (see p. 195). The flow of steam, however, through the calorimeter must always be started before observations of the temperatures are to be taken in order to get **constant conditions**.

¹ PULLEN, W. W. F., “Engines and Boilers,” pp. 466–475.

² Economizer and air heater are not included when the heat absorbed is not returned to the unit.

Boiler testing should not be undertaken by anyone who has not had some training under an experienced testing engineer if reliable results are to be expected. The whole subject should be thoroughly understood, both theoretically and practically. Accurate tests depend very largely upon the care and faithfulness of the observers. It is much easier to make mistakes than is realized by those who are not familiar with practical testing.

The limits of accuracy of a test may very reasonably be taken to be within plus or minus 3 per cent. One of the sources of probable error is the sampling of coal. Even when the greatest care is taken to obtain a representative sample, there may be an indeterminate error in ascertaining the heating value of the coal, even though the laboratory analysis is most reliable. With modern apparatus these laboratory determinations should be substantially correct as regards the sample tested; but the question as to how truly the sample represents the whole is always present and cannot be answered definitely.

Another source of error is the moisture contained in the coal. The sampling is more or less uncertain. It is contended by some that, if the attempt is made to determine the moisture during the test, the methods of drying and weighing are unreliable; while others contend that, though the moisture as determined in the laboratory is accurate as far as the sample delivered to the laboratory is concerned, this sample probably does not represent the bulk of the coal actually burned, since there must inevitably have been more or less loss of moisture during the collection, preparation, and handling of the sample.

Similarly, it is problematical whether the samples collected for the determination of the moisture in steam and for gas analysis are representative of the bulk, although the testing of the samples obtained may be quite accurate.

It is not unusual for heat balances to be reported to the nearest B.t.u. and to the nearest $\frac{1}{10}$ per cent. But the present state of the art of boiler testing does not provide means for attaining anything like this accuracy. In general, results should be reported only to the nearest significant figure.

It is therefore quite logical in the case of *guarantee tests* that a substantial compliance with the guarantee be accepted as full compliance therewith.

Measurements.—The quantities listed below are the principal ones which must be measured in a performance test of a stationary steam boiler.

(a) Area of heating surfaces; (b) grate surface (p. 303, footnote); (c) furnace volume (p. 303, footnote) and dimensions; (d) analysis and heating value of fuel; (e) average steam pressure; (f) quality of steam; (g) analysis of flue gases; (h) draft pressures at various points; (i) steam temperatures; (j) superheat; (k) various air

temperatures; (*l*) gas temperatures; (*m*) feed-water temperatures; (*n*) total weight of fuel; (*o*) total weight of refuse; (*p*) analysis of refuse; (*q*) weight of water.

Instruments and apparatus required for boiler tests are:

(*a*) Scales for weighing coal, oil, or other fuel, ashes, furnace refuse, etc.; (*b*) graduated scales attached to the water glasses; (*c*) tanks or tanks and scales for volumetric or weight measurement of water; (*d*) meters or other apparatus for measuring gaseous fuels; (*e*) pressure and draft gages; (*f*) thermometers and pyrometers; (*g*) calorimeter for determining the quality of the steam; (*h*) gas-sampling and analyzing apparatus.

Weighing Fuel.—Fuel-weighing apparatus should be near the point where the fuel is to be used, under direct observation of and convenient to the one in charge of the test. The weighing of refuse may be done at any convenient point.

Water Glass.—The water-glass scale should be so attached that the breaking of the glass will not disturb the scale.

Weighing Water.—The water-weighing or measuring apparatus if other than a meter should be arranged in an easily accessible place and made as convenient as practicable. It should also be under the direct observation of the one in charge of the test.

Gas Meters.—Meters or other apparatus for measuring gaseous fuel may be placed in such locations as conditions dictate.

Gages.—Pressure gages should be away from any disturbing influences, such as extreme heat, and in a position to be easily read.

Draft should be measured at not less than two points: (*a*) after the flue gas has passed beyond contact with any part of the boiler and before it has passed the damper, or before it has entered the outlet duct if there is no damper; and (*b*) in the furnace immediately before the flue gas comes in contact with the heating surface of the boiler proper, which in the case of vertical- or steep-tube types of water tube boilers should be midway of the tube length.

Draft at the gas outlet is usually measured if nowhere else. It is often desirable to know the draft at other points of the setting, in which case the several locations should be chosen to suit the information wanted. Care should be exercised to see that the ends of the connecting pipes are not obstructed and that the open ends are not subject to other than static pressures. All joints of draft-gage connections must be tight.

Thermometers.—A thermometer well for measuring feed-water temperatures should be as close to the boiler inlet as possible and the pipe between boiler and thermometer should be well protected with heat-insulating covering.

Saturated-steam temperatures may be measured at any point in the steam pipe where the pressure is the same as that where the temperature is desired, care being taken that the well is not cooled by the condensate. The temperature and pressure of superheated steam should be measured as close to the outlet of the superheater as possible, with all losses by pressure drop and radiation between the superheater and thermometer eliminated.

Pyrometers of any type must have the part on which the heat impinges so located that the temperature which it is desired to measure is actually obtained. Much experience and good judgment are essential in order to obtain even approximately reliable readings. Thermometers for determining atmospheric temperatures should be placed so as to give an average indication away from cold or hot drafts, etc.

Steam Calorimeter.—The sampling tube of the steam calorimeter should be as close as possible to the point at which the quality of steam is desired. All exposed piping in which condensation might influence the result must be well lagged. Ordi-

narily the quality of the steam leaving the boiler, or that entering the superheater, is desired.

Flue-gas Analysis.—Analysis of the exit gases is usually what is taken, but very frequently analyses are wanted from other points. Sample tubes must be so located that only the gas to be analyzed enters the tube and precautions taken against the possibility of inleakage of air anywhere in the line. The gas-analyzing apparatus should be in a readily accessible and protected place with good natural or artificial light and convenient facilities.

Determine the character of the fuel to be used. If it is to be coal and its size is to be reported, it should be determined in accordance with the Test Code for Solid Fuels (see page 259).

Starting and Stopping.—Combustion, fuel, draft, and temperature conditions, the water level, rate of feeding water, rate of steaming, and the steam pressure should be as nearly as possible the same at the end as at the beginning of the test. If an economizer is included in the unit tested, the average temperature of the water within it should be the same at the start and end of test.

Hand Firing.—To secure the desired equality of conditions with hand-fired boilers, the following method should be employed:

The furnace being well heated by a preliminary run at the same combustion rate that will prevail during the test and sufficiently long to heat the setting thoroughly, burn the fire low and thoroughly clean it, leaving enough live coal spread evenly over the grate (say, from 2 to 4 inches) to serve as a foundation for the new fire. Note quickly the thickness of the coal bed as nearly as it can be estimated or measured, also the water level, the steam pressure, and the time, and record the last as the starting time. Fresh coal should then be fired from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level, the steam pressure, and the time, and record the last as the stopping time. If the water level is lower or higher than at the beginning, a correction should be made by computation, not by feeding additional water. Finally remove the ashes and refuse from the ashpit.

In a plant containing several boilers, where it is not practicable to clean them simultaneously, the fires should be cleaned one after the other as rapidly as may be, and each one after cleaning charged with enough coal to maintain a thin fire in good working condition.

After the last fire is cleaned and in working condition, burn all the fires low (say, 4 to 6 inches), note quickly the thickness of each, also the water levels, steam pressure, and time, which last is taken as the starting time. Likewise, when the time arrives for closing the test, the fires should be quickly cleaned one by one, and when this work is completed they should all be burned low the same as at the start, and the various final observations made as before stated.

In the case of a large boiler having several furnace doors requiring the fire to be cleaned in sections one after the other, the foregoing directions pertaining to starting and stopping in a plant of several boilers may be followed.

Mechanical Stokers.—To obtain the desired equality of conditions when a mechanical stoker is used, the foregoing procedure should be modified as follows:

Stokers should be in operation at approximately the same rate that will prevail during the test for at least 12 hours before starting the test. At the start and finish of the test, level the coal in the stoker hopper. Make starting and stopping observations as in hand-fired tests.

Stokers of the Continuous-dumping Type.—The desired operating conditions, that is, speed and stroke of coal-feeding mechanism, speed of grate, intensity of draft or blast, and rate of water feed, should be maintained as nearly constant as possible for at least 1 hour, and preferably for 2 hours, before starting and stopping the test.

Stokers of the Intermittent-dumping Type.—Proceed as above, except that stokers should be cleaned about 1 hour before starting and before stopping the test.

To obtain the desired equality of conditions when pulverized, liquid, or gaseous fuel is used, the following method should be employed: The boiler or boilers to be tested should be operated before the start of the test under the fuel, furnace, and combustion conditions which are to be maintained throughout the test for a period of not less than 3 hours. Fuel temperature, fuel pressure, and draft conditions should be kept as nearly constant as possible during this period and throughout the test. The references above for hand-fired tests as to starting and stopping observations hold also for pulverized-, liquid-, and gaseous-fuel tests.

Duration.—The duration of tests to determine the efficiency of a steam-generating unit burning **solid fuel** either hand or stoker fired should preferably be 24 hours; but where operating conditions do not permit or other considerations make it advisable or necessary, the length of test may be reduced to not less than 10 hours. When the rate of combustion is less than 25 pounds of coal per square foot of grate surface per hour, the test should be continued until a total of 250 pounds per square foot of grate area has been burned, except that, in cases where a type of stoker is used that does not permit the quantity of fuel and the condition of the fuel bed to be accurately estimated, the duration of test should not be reduced below that required to minimize the error.

In a unit using pulverized fuel the duration of test should be not less than 6 hours, and for liquid or gaseous fuel not less than 4 hours.

In the case of a waste-heat unit, where the operation of the industrial furnace is continuous and furnace conditions are constant, the duration of the test should be not less than 6 hours. Where the industrial-furnace operation is in cycles, the test should be of such duration as to cover at least one cycle of furnace operation.

In tests conducted under plant-operation conditions where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of test should be at least 24 hours. Likewise in such tests, either of a single unit or of a plant of several units which operate regularly a certain number of hours and during the remainder of the day are banked, the duration should be not less than 24 hours.

The duration of tests to determine the maximum evaporative capacity of a steam-generating installation, when the efficiency is not determined, should be not less than 2 hours, unless otherwise agreed upon.

Records.—The data listed in the tables of this code should be obtained and recorded in convenient form. Readings of the instruments at 15-minute intervals are usually sufficient. If there are sudden and wide fluctuations, the readings should be taken at 10-minute intervals or at such frequency as may be necessary to determine a good average.

Fuel.—The approximate quantity of fuel needed each hour should be determined; and if the quantity required and inconvenience due to lack of room are not too great, the whole amount should be delivered on the firing floor at the beginning of each hour. If any is left at the end of the hour, the quantity should be estimated. In this way the amount of fuel used per hour can be determined and the approximate results be followed from hour to hour.

If the whole amount cannot be delivered at the beginning of the hour, convenient quantities may be weighed out at appropriate intervals so as to come out about even at the end of the hour. When hopper scales are used, receiving coal from bunkers and discharging directly into furnace hoppers, hourly quantities may be roughly determined by estimating furnace conditions. These hourly quantities should be properly noted on the log sheet. They are taken as a matter of convenience and guide, but only the totals are to be used in the final calculations.

Steam Rate.—The records should be such as to ascertain also the approximate consumption of feed water each hour and thereby determine the degree of uniformity of evaporation. The maintenance of a uniformity of evaporation is greatly facilitated by the use of some form of graphic recording steam meter, which should be so placed as to keep continuously before the operator the rate of evaporation. Since the indications of this meter are not used in any of the test calculations, extreme accuracy is not essential.

Quality of Steam.—If the boiler does not produce superheated steam, the quality of the steam should be determined by the use of a throttling or separating calorimeter. If the boiler has superheating surface, the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer well or by a suitable pyrometer.

Sampling Fuel.—During the progress of the test, fuel should be regularly sampled for purposes of analysis.

Ashes and Refuse.—The ashes and refuse withdrawn from the furnace and ashpit during the progress of and at the end of the test should be weighed, as far as possible, in a dry state. If wet, the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may also serve for analysis.

Ash sampling is at best subject to large errors, and every precaution should be taken to insure as representative a sample as possible. If sufficiently hot to allow combustion to proceed, ash should be thoroughly quenched with water immediately

after dumping into the ashpit. Enough time should elapse before weighing to permit the heat of the refuse to drive out most of this moisture. If possible, all the ash should then be passed through a crusher, thoroughly mixed, and reduced to a laboratory-size sample by successive quartering (page 262). If it is impracticable to crush all the ash, the clinkers and "fines" should be placed in separate piles, and each pile weighed and sampled separately, the two samples being combined in proportion to the relative weights of their respective piles.

Calorific Tests and Analyses of Fuels and Refuse.—The quality of the fuel and of the furnace ash and refuse should be determined by calorific tests and analyses of the samples above referred to. The *higher heat value* (page 242) is used throughout this code. Directions for obtaining samples of the fuel and making these tests and analyses are given on pages 222 and 258.

Flue-gas Analyses.—For approximate determinations of the composition of the flue gases, the Orsat apparatus or some modification thereof should be employed. Gas samples should preferably be taken continuously; but if momentary samples are obtained, the analyses should be made as frequently as possible, the furnace and firing conditions being noted at the time the samples are drawn. Where the firing is intermittent, gas samples should be taken at such intervals that the complete firing cycle will be covered by the average of individual readings. If the sample drawn is a continuous one, the intervals may be made longer. Where the unit includes an economizer, gas analyses should be made at both boiler and economizer outlets.

Smoke Observations.—In tests requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of 5 minutes or, if necessary, of 1 minute. For observations covering a period of one or more single firings with solid fuel, the intervals should be $\frac{1}{4}$ minute or less.

No wholly satisfactory methods for either quantitative or qualitative smoke determinations have yet come into use, nor have any reliable methods been established for definitely fixing even the relative density of the smoke issuing from chimneys at different times. One method commonly employed, which answers the purpose fairly well, is that of making frequent visual observations of the chimney at intervals of 1 minute or less for a period of 1 hour and recording the observed characteristics according to the degree of blackness and density, and giving to the various degrees of smoke an arbitrary percentage value rated in some such manner as the following:

SMOKE PERCENTAGES			
Dense black.....	100	Light gray.....	20
Medium black.....	80	Very light.....	5
Dense gray.....	60	Trace.....	1
Medium gray.....	40	Clear chimney.....	0

The color and density of smoke depend somewhat on the character of the sky or other background and on the air and weather conditions obtaining when the observation is made, and these should be given due consideration in making comparisons. Observations of this kind are also subject to personal errors and errors of judgment.

Nevertheless, these methods are useful especially when the results are plotted, according to the percentage scale determined on, so that a graphic representation of the changes can be shown. The observations should be considered as only roughly indicating the smoke conditions and are comparable only when made by the same observer.

Various forms of charts and clouded-glass arrangements for comparing and fixing smoke densities have been proposed, and to some extent used, but these have proved more or less unsatisfactory and they are subject to personal errors and to sky, wind, and weather conditions, the same as the simpler method above described.

Power for Auxiliaries.—It is recommended that a record be kept of the energy used by auxiliaries immediately connected with the steam-generating unit being tested. This applies particularly to steam or power used in driving stokers or other fuel-feeding apparatus, fans, feed pumps, and soot blowers and may also include energy used in fuel preparation.

It is always difficult and usually impractical to determine accurately the steam equivalent of the power used by auxiliaries. For this reason no deduction is made in computing Items 93 to 97 (evaporation units), Items 104 and 105 (efficiency), or any data of the heat balance unless required by the object of the test. It is frequently desirable, however, to compare the performance of installations of different types, and for this purpose a close approximation to the overall efficiency of the steam-generating unit, corrected for auxiliaries, is of value. The deduction of such approximate auxiliary-energy steam equivalent from the total steam will give an approximate "net efficiency of the steam-generating unit." The Code makes no attempt to define this item exactly or to specify how it should be determined, because of essential dependence upon the characteristics of the individual installation. Whenever this "net" value is given, a full statement should accompany the report, stating the apparatus for which deductions have been made and the method used in determining the deductions.

Graphical Log for Report.—In trials having for an object the determination and exposition of the complete boiler performance, the entire log of readings and data should be represented graphically in the report.

Calculation of Results.—All calculations incident to a test of a stationary steam-generating unit are outlined in detail in the following tables. These tables are divided into three main groups lettered *S*, *L*, and *G*, symbols which stand for the *type of fuel* used, solid, liquid, and gaseous, respectively. In each there are numbered subdivisions indicating the appropriate computations for the various combinations of auxiliary apparatus.

TABLE S1A.—DATA AND RESULTS OF STATIONARY STEAM-GENERATING UNIT TEST.
SOLID FUELS—AS FIRED—DRY

General Information

(1) Date of test; (2) location; (3) owner; (4) maker and type of boiler; (5) maker and type of water walls; (6) maker and type of superheater; (7) maker and type of

economizer; (8) maker and type of reheater; (9) maker and type of air heater; (10) maker and type of fuel-burning equipment; (11) test conducted by; (12) object of test.

Description, Dimensions, Etc.

13. Boiler-heating surface.....	sq. ft.
14. Water-wall surface (water walls, screen or floor, and roof).....	sq. ft.
15. Superheater surface.....	sq. ft.
16. Economizer surface.....	sq. ft.
17. Reheater surface.....	sq. ft.
18. Total steam-generating unit surface.....	sq. ft.
19. Air-heater surface.....	sq. ft.
20. Grate surface.....	sq. ft.
21. Fuel-burning equipment.....	
22. Method of producing draft.....	
23. Fuel.....	
24. Volume of combustion space.....	cu. ft.
25. Furnace, center of grate to nearest heating surface.....	ft.
26. Furnace volume per square foot boiler-heating surface.....	cu. ft.

Fuel and Gas Analyses and Data

Fuel Proximate Analysis—As Fired—Dry

27. Volatile matter.....	per cent
28. Fixed carbon.....	per cent
29. Ash.....	per cent
30. Moisture (as fired).....	per cent
31. Heating value per pound (as fired).....	B.t.u.
32. Heating value per pound (dry).....	B.t.u.
33. Fusion temperature of ash.....	°F.
34. Size of coal as fired.....	

Fuel Ultimate Analysis—As Fired—Dry

35. Carbon.....	per cent
36. Hydrogen.....	per cent
37. Oxygen.....	per cent
38. Nitrogen.....	per cent
39. Sulphur.....	per cent
40. Ash.....	per cent
40a. Moisture.....	per cent

Gas

41. Gas analysis, furnace:	
Per cent	CO ₂ O ₂ CO.....N ₂ SO ₂
42. Gas analysis, boiler outlet:	
Per cent	CO ₂ O ₂ CO.....N ₂ SO ₂
43. Gas analysis, economizer outlet:	
Per cent	CO ₂ O ₂ CO.....N ₂ SO ₂
44. Gas analysis, air-heater outlet:	
Per cent	CO ₂ O ₂ CO.....N ₂ SO ₂
45. Dry gas per pound of fuel, in furnace [(a) as fired, (b) dry].....	lb.
46. Dry gas per pound of fuel, at boiler outlet [(a) as fired, (b) dry].....	lb.
47. Dry gas per pound of fuel, at economizer outlet [(a) as fired, (b) dry].....	lb.
48. Dry gas per pound of fuel, at air-heater outlet [(a) as fired, (b) dry].....	lb.

49. Dry gas per pound of fuel, theoretical [(a) as fired, (b) dry].....lb.
 50. Air supplied per pound of fuel, in furnace [(a) as fired, (b) dry].....lb.

Pressures and Drafts

51. Moisture in air.....lb. per lb. air
 52. Steam pressure by gage, at boiler.....lb. per sq. in.
 53. Steam pressure by gage, at superheater outlet.....lb. per sq. in.
 54. Steam pressure by gage, at reheater inlet.....lb. per sq. in.
 55. Steam pressure by gage, at reheater outlet.....lb. per sq. in.
 56. Air pressure in ashpit zone, at burners.....in. of water
 57. Draft in furnace.....in. of water
 58. Draft at boiler outlet.....in. of water
 59. Draft at economizer outlet.....in. of water
 60. Draft at air-heater outlet.....in. of water

Temperatures

61. Steam temperature at superheater outlet.....°F.
 62. Moisture in steam.....per cent
 63. Superheat.....°F.
 64. Moisture or superheat in steam entering reheater.....per cent or ... °F.
 65. Superheat in steam leaving reheater.....°F.
 66. Temperature of air surrounding boiler (t_1).....°F.
 67. Temperature of air entering air heater (t_2).....°F.
 68. Temperature of air leaving air heater (t_3).....°F.
 69. Temperature of air for combustion (t_4).....°F.
 70. Temperature of furnace (t_5).....°F.
 71. Temperature of gases leaving boiler (t_6).....°F.
 72. Temperature of gases leaving economizer (t_7).....°F.
 73. Temperature of gases leaving air heater (t_8).....°F.
 74. Temperature of feed water entering boiler (t_9).....°F.
 75. Temperature of feed water entering economizer (t_{10}).....°F.
 76. Temperature of water in boiler at point where gases leave boiler (t_{11}).....°F.
 77. Temperature of fuel (t_{12}).....°F.

Hourly and Unit Quantities

78. Duration of test.....hr.
 79. Fuel as fired per hour.....lb.
 80. Dry fuel per hour.....lb.
 81. Fuel as fired per square foot of grate per hour.....lb.
 82. Fuel as fired per retort or per burner per hour.....lb.
 83. Dry fuel per square foot of grate per hour.....lb.
 84. Dry fuel per retort or per burner per hour.....lb.
 85. Combustion space per pound of coal per hour [(a) as fired, (b) dry].....cu. ft.
 86. Refuse per hour.....lb.
 87. Actual water evaporated per hour.....lb.
 88. Steam through reheater per hour.....lb.
 89. Heat absorbed by water in economizer.....B.t.u. per lb.
 90. Heat absorbed by water and steam in boiler.....B.t.u. per lb.
 91. Heat absorbed by steam in superheater.....B.t.u. per lb.
 92. Heat absorbed by steam in reheater.....B.t.u. per lb.
 93. Rate of heat absorption in economizer.....kB* per hr.
 94. Rate of heat absorption in boiler.....kB per hr.
 95. Rate of heat absorption in superheater.....kB per hr.

96. Rate of heat absorption in reheater.....*kB* per hr.
 97. Total rate of heat absorption by steam-generating unit.....*kB* per hr.

Refuse

98. Refuse, per cent of fuel [(a) as fired, (b) dry].....per cent
 99. Percentage of combustible in refuse.....per cent
 100. Carbon burned per pound fuel [(a) as fired, (b) dry].....lb.

Evaporation

101. Rate of heat absorption per pound fuel (as fired).....*kB**
 102. Rate of heat absorption per pound fuel (dry).....*kB*
 103. Rate of heat absorption per square foot of steam-generating unit surface
 per hour.....*kB*

Efficiency

104. Efficiency of steam-generating unit.....per cent
 105. Comparative efficiency of steam-generating unit (when not equipped
 with air heater).....per cent

* For explanation of this symbol see p. 299.

TABLE S1B.—HEAT BALANCE OF STATIONARY STEAM-GENERATING UNIT. SHORT FORM. SOLID FUELS

	B.t.u. per Lb. Fuel, as Fired, Dry*	Per Cent
1105. Heating value of fuel.....		
1106. Heat absorbed by water in economizer.....		
1107. Heat absorbed by water and steam in boiler.....		
1108. Heat absorbed by steam in superheater.....		
1109. Heat absorbed by steam in reheater.....		
1109a. Heat absorbed by steam-generating unit.....		
1110. Heat loss due to moisture in coal.....		
1111. Heat loss due to water from combustion of hydrogen....		
1112. Heat loss due to moisture in air.....		
1113. Heat loss due to dry chimney gases.....		
1114. Heat loss due to incomplete combustion of carbon.....		
1115. Heat loss due to unconsumed combustible in refuse....		
1116. Heat loss due to unconsumed hydrogen and hydrocar- bons, radiation, and unaccounted for.....		

* If test is on a dry basis, draw a line through "as fired" wherever the words "as fired, dry" appear together, and *vice versa*. Whichever basis is chosen must be followed throughout. Percentage results in heat balance will be the same in either case.

Explanation of Computations for Table S1A

Wherever CO₂, O₂, CO, and N₂ are used, they are the percentages by volume of these constituents in the gases of combustion.

h_g = total heat (in B.t.u. per pound) of saturated steam at the boiler outlet pressure. See tables in Appendix.

h_{g1} = total heat (in B.t.u. per pound) of superheated steam at the superheater outlet pressure. See tables in Appendix.

h_{g2} = total heat (in B.t.u. per pound) of the steam at the reheater inlet pressure.

h_{gs} = total heat (in B.t.u. per pound) of superheated steam at the reheater outlet pressure.

h_{gi} = total heat (in B.t.u. per pound) in feed water at boiler inlet.

h_{po} = total heat (in B.t.u. per pound) in feed water at economizer inlet.

h_{fg} = heat of evaporation (in B.t.u. per pound) in steam at pressure in steam main.

t_s = temperature of steam after expansion in calorimeter.

t_{13} = initial temperature of water for quenching ashes.

t_{14} = final temperature of water for quenching ashes.

W = weight of water used per hour for quenching ashes.

kB = absorption of 1000 B.t.u. (see p. 299).

If test is on a dry basis, draw a line through "as fired, dry" wherever the words appear together, and *vice versa*. Whichever basis is chosen must be followed throughout. Percentage results in heat balance will be the same in either case.

Explanation of Items Already Listed in Table S1A. Numbers in Margin Refer to Items of the Same Number

13 and 14. Boiler heating surface shall consist of that portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor.

Heating surface located in the furnace or furnace boundaries shall be measured as follows:

Tubes wholly exposed or wholly or partly embedded in refractory—that portion of the surface of the tubes which is exposed to the gas or refractory being cooled.

Tubes provided with extended surface—that portion of the tubes and their extensions which is exposed to the gas being cooled.

Tubes protected by blocks rigidly attached to the fire faces—that portion of the surface of the blocks which is exposed to the gas being cooled.

15. Superheater surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to steam being heated after leaving the boiler circulating system, the surface being measured on the flue-gas side.

16. Economizer surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to fluid being heated before entering the boiler circulating system, the surface being measured on the flue-gas side.

17. Reheater surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to steam being heated after expansion, the surface being measured on the flue-gas side.

18. The steam-generating unit heating surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas or refractory being cooled and on the other to the fluid being heated, measured on the side receiving heat. This item equals the sum of Items 13, 14, 15, 16, and 17.

19. Air-heater surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to air being heated for combustion, the surface being measured on the flue-gas side except that in air heaters of the regenerative type, where the surfaces are alternately in contact with gas and air, the air-heater surface shall be determined for the purpose of this code by multiplying the total surface of the apparatus by that proportion continuously in contact with the gases being cooled.

20. The horizontal projected area within the four walls surrounding the furnace measured at the middle of the grate.

22. State whether natural, forced, induced, or a combination of one or more kinds of draft was employed.

23. State general class of fuel, as anthracite, bituminous, wood refuse, etc. Also give district where produced.

24. The cubic space provided for the combustion of fuel before the products of combustion pass through any heating surface.

25 to 40. "As fired" analysis items = dry analysis items $\times \left(1 - \frac{\text{Item 30}}{100}\right)$.

45 to 48. $\frac{11\text{CO}_2 + 8\text{O}_2 + 7(\text{CO} + \text{N}_2)}{3(\text{CO}_2 + \text{CO})} \times \text{Item 100}$. See Eq. (76) (p. 330).

Note. In the following equations the percentage of carbon is multiplied by 12.52, the percentage of hydrogen by 26.56, the percentage of sulphur by 5.325, and the percentage of nitrogen by 1.0 to obtain the dry gas per pound of fuel.

49. $12.52 \times \text{Item 100} + \frac{26.56 \times \text{Item 36} + 5.325 \times \text{Item 39} + \text{Item 38}}{100}$.

50. $\text{Item 45} + \frac{9 \times \text{Item 36}}{100} - \frac{100 - \text{Item 98}}{100}$.

62. Determine either from charts or if throttling calorimeter expanding to atmospheric pressure is used, by the following formula:

$$100 \times \frac{h_g - 1,150.4 - C_p \dagger \times (t_s - 212)}{h_{fg}}$$

86. Combined weight of ashes and cinders produced during test divided by Item

69. In tests of large boilers where it is impracticable to determine the amount of refuse by weighing, Item 77 may be calculated as follows:

$$\text{Item 86} = \frac{\text{Item 29}}{100 - \text{Item 99}} \times \text{Item 79 (or 80)}.$$

$$89. = (h_{gi} - h_{go}).$$

$$90. = (h_g - h_{gi}).$$

$$91. = (h_{g1} - h_g).$$

$$92. = (h_{g3} - h_{g2}).$$

$$93. = \frac{\text{Item 87} \times \text{Item 89}}{1,000}$$

$$94. = \frac{\text{Item 87} \times \text{Item 90}}{1,000}$$

$$95. = \frac{\text{Item 87} \times \text{Item 91}}{1,000}$$

$$96. = \frac{\text{Item 88} \times \text{Item 92}}{1,000}$$

$$97. = \text{Item 93} + \text{Item 94} + \text{Item 95} + \text{Item 96}$$

$$98. = \frac{\text{Item 86}}{\text{Item 79 (or 80)}}$$

* The percentage of hydrogen is multiplied by 9. See Item 1111 of this code.

† As shown by the curves of mean values of the specific heat of superheated steam on p. 369, the approximate value at atmospheric pressure is 0.46. For the calculation of this item in the Code, C_p is given the value 0.47, presumably to be an average value to cover also cases where the calorimeter pressure is slightly above atmospheric. Whether the accepted value is 0.46 or 0.47 makes very little difference in results.

$$\begin{aligned}
 100. &= \frac{\text{Item 35}}{100} - \frac{\text{Item 98} \times \text{Item 99}}{100 \times 100} \\
 101. &= \frac{\text{Item 97}}{\text{Item 79}} \\
 102. &= \frac{\text{Item 97}}{\text{Item 80}} \\
 103. &= \frac{\text{Item 97}}{\text{Item 18}} \\
 104. &= \frac{\text{Item 101 (or 102)} \times 1,000 \times 100}{\text{Item 31 (or 32)}} \\
 105. &= \frac{\text{Item 101} \times 1,000 \times 100}{\text{Item 31} + (t_4 - 70) \times (\text{Item 50} \times 0.24)^*}
 \end{aligned}$$

or

$$\frac{\text{Item 102} \times 1,000 \times 100}{\text{Item 32} + (t_4 - 70) \times (\text{Item 50} \times 0.24)^*}$$

Explanation of Computations for Table S1B

$$\begin{aligned}
 1106. &= \frac{\text{Item 93} \times 1,000}{\text{Item 79 (or 80)}} \\
 1107. &= \frac{\text{Item 94} \times 1,000}{\text{Item 79 (or 80)}} \\
 1108. &= \frac{\text{Item 95} \times 1,000}{\text{Item 79 (or 80)}} \\
 1109. &= \frac{\text{Item 96} \times 1,000}{\text{Item 79 (or 80)}} \\
 1109a. &= \text{Item 1106} + \text{Item 1107} + \text{Item 1108} + \text{Item 1109} \\
 1110. &= \frac{\text{Item 30}}{100} \times (1090.7 + 0.455 t_6 - t_{12}).\dagger \\
 1111. &= \frac{\text{Item 36}}{100} \times 9\dagger \times (1090.7 + 0.455 t_6 - t_{12}). \text{ If economizer is installed}
 \end{aligned}$$

without air heater, substitute t_7 for t_6 ; if air heater is installed, substitute t_8 for t_6 in Items 1110, 1111, 1112, and 1113.

1112. = Item 50 $\times$ Item 51 $\times$ 0.47 ($t_6 - t_1$). This loss is small and is frequently included in Item 1113.

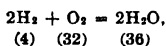
1113. = Item 46 (47 or 48) $\times$ 0.24 ($t_6 - t_1$). If boiler and superheater, use Item 46; if economizer, use Item 47; if air heater, use Item 48. Use data from Item 42, 43, or 44, depending on whether boiler and superheater are installed alone; or with economizer but without air heater, or with air heater.

1114. = $\frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \text{Item 100} \times 10,160.\P$ If boiler alone, use analysis from Item 42; if economizer, use Item 43; if air heater, use Item 44.

* The value in common use for the specific heat of dry flue gases is 0.24.

† For explanation of temperature symbols see p. 314. The specific heat of the gases leaving the boiler is assumed to be 0.455. The total heat of steam for the conditions stated is 1090.7 B.t.u.

‡ Weight of moisture (H_2O) from combustion of hydrogen (H_2) in fuel is nine times weight of hydrogen, thus



which shows that the molecular weight of the moisture is 36 $\div$ 4 or 9 times that of the hydrogen.

\P See footnote * on page 319.

$$1115. = \frac{\text{Item 98} - \text{Item 29}}{100} \times 14,600.*$$

$$1116. = \text{Item 31 (or 32)} - (\text{sum of Items 1106 to 1115 inclusive}).$$

* Heat of combustion of a pound of carbon burned to carbon dioxide is 14,600 B.t.u., and of a pound of carbon burned to carbon monoxide is 4,440 B.t.u. Heat lost by burning carbon to carbon monoxide (incomplete combustion) is 14,600 - 4,440) or 10,160 B.t.u. per pound of carbon.

TABLE S1C.—HEAT BALANCE OF STEAM-GENERATING UNIT COMPRISING BOILER, SUPERHEATER, AND REHEATER, WITH OR WITHOUT INTEGRAL ECONOMIZER. SOLID FUELS

	<i>Heat Values</i>	
	B.t.u. per Pound Fuel as Fired, Dry	Per Cent
2106. Heating value of fuel (higher heating value).....		
2107. Heat absorbed by water and steam in boiler.....		
2108. Heat absorbed by steam in superheater.....		
2109. Heat absorbed by steam in reheater.....		
2109a. Heat absorbed by steam-generating unit.....		
<i>Unavoidable Losses</i>		
2110. Heat loss due to moisture in coal, moisture in theoretical air and water from combustion of hydrogen, up to t_{11}		
2111. Heat loss due to theoretical dry gases, t_4 to t_{11}		
2112. Total unavoidable losses.....		
<i>Other Losses</i>		
2113. Heat loss due to combustible in refuse.....		
2114. Heat loss due to sensible heat in refuse, cinders, and soot.....		
2115. Heat loss due to unburned gaseous combustibles.....		
2116. Heat loss due to excess air entering furnaces and mois- ture accompanying same, t_4 to t_{11}		
2117. Heat loss due to theoretical dry gases, moisture in coal, moisture accompanying theoretical air, and moisture from combustion of hydrogen, t_{11} to t_6		
2118. Heat loss due to excess air entering furnace and moisture accompanying same, t_{11} to t_6		
2119. Heat loss due to air and moisture leaking through boiler setting, t_1 to t_6		
2120. Heat loss due to unconsumed hydrogen and hydrocar- bons, radiation, and unaccounted for.....		

Explanation of Computations for Table S1C

$$2106. = \text{Item 31 (or 32)}.$$

$$2107. = \frac{\text{Item 94} \times 1,000}{\text{Item 79 (or 80)}}.$$

$$2108. = \frac{\text{Item 95} \times 1,000}{\text{Item 79 (or 80)}}.$$

$$2109. = \frac{\text{Item 96} \times 1,000}{\text{Item 79 (or 80)}}.$$

$$2109a. = \text{Item 2107} + \text{Item 2108} + \text{Item 2109}.$$

2110. = $(1,090.7 + 0.455 t_{11} - t_{12}) \times \frac{(\text{Item 30} + 9* \times \text{Item 36})}{100}$
 + $(\text{Item 49} - \text{Item 100}) \times \text{Item 51} \times 0.47 \times (t_{11} - t_4)$.
 2111. = $\text{Item 49} \times (t_{11} - t_4) \times 0.24$.†
 2112. = $\text{Item 2110} + \text{Item 2111}$.
 2113. = $\frac{\text{Item 98} - \text{Item 29}}{100} \times 14,600$.
 2114. = $\frac{W \times (t_{14} - t_{13})}{\text{Item 79 (or 80)}}$. This item can be determined only in plants where ash and refuse are discharged into water.
 2115. = $\frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \text{Item 100} \times 10,160$ (use data from Item 42).
 2116. = $(\text{Item 45} - \text{Item 49}) \times (t_{11} - t_4) \times (0.24 + \text{Item 51} \times 0.47)$.‡
 2117. = $(t_6 - t_{11}) \times \left\{ \text{Item 49} \times 0.24 + [(\text{Item 49} - \text{Item 100}) \times \text{Item 51} + \frac{\text{Item 30} + 9 \times \text{Item 36}}{100}] \times 0.47 \right\}$.
 2118. = $(\text{Item 45} - \text{Item 49}) \times (t_6 - t_{11}) \times (0.24 + \text{Item 51} \times 0.47)$.
 2119. = $(\text{Item 46} - \text{Item 45}) \times (t_6 - t_1) \times (0.24 + \text{Item 51} \times 0.47)$.
 2120. = $\text{Item 2106} - \text{sum of Items 2107, 2108, 2109, 2110, 2911, and 2113 to 2119 inclusive}$.

* See footnote, p. 318.

† This is the value in common use for the specific heat of dry flue gases.

‡ The items in detail before combination are:

$$(\text{Item 45} - \text{Item 49}) \times 0.24 \times (t_{11} - t_1) + (\text{Item 45} - \text{Item 49}) \times \text{Item 51} \times 0.47 \times (t_{11} - t_4).$$

TABLE S3B.—HEAT BALANCE OF STEAM-GENERATING UNIT COMPRISING BOILER, SUPERHEATER, REHEATER AND ECONOMIZER. SOLID FUELS

Heat Values

	B.t.u. per Pound Fuel as Fired, Dry	Per Cent
3106. Heating value of fuel (higher heating value).....		
3107. Heat absorbed by water in economizer.....		
3108. Heat absorbed by water and steam in boiler.....		
3109. Heat absorbed by steam in superheater.....		
3110. Heat absorbed by steam in reheater.....		
3110a. Heat absorbed by steam-generating unit.....		

Unavoidable Losses

3111. Heat loss due to moisture in coal, moisture accompanying theoretical air, and water from combustion of hydrogen, up to t_{10}		
3112. Heat loss due to theoretical dry gases, t_4 to t_{10}		
3113. Total unavoidable losses.....		

Other Losses

3114. Heat loss due to combustible in refuse.....	
3115. Heat loss due to sensible heat in refuse, cinders, and soot	
3116. Heat loss due to unburned gaseous combustibles.....	
3117. Heat loss due to excess air entering furnace and moisture accompanying same, t_4 to t_{10}	

3118.	Heat loss due to theoretical dry gases, moisture in coal, moisture accompanying theoretical air and water from combustion of hydrogen, t_{10} to t_7		
3119.	Heat loss due to excess air entering furnace and moisture accompanying same, t_{10} to t_7		
3120.	Heat loss due to air and moisture leaking through boiler setting, t_1 to t_7		
3121.	Heat loss due to air and moisture leaking through economizer setting, t_1 to t_7		
3122.	Heat loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted for.....		

Economizer

3123.	Heat available to economizer in flue gases including moisture from t_6 to t_{10}		
3124.	Efficiency of economizer.....		

Explanation of Computations for Table S3B

3106.	= Item 31 (or 32).
3107.	= $\frac{\text{Item 93} \times 1,000}{\text{Item 79 (or 80)}}$.
3108.	= $\frac{\text{Item 94} \times 1,000}{\text{Item 79 (or 80)}}$.
3109.	= $\frac{\text{Item 95} \times 1,000}{\text{Item 79 (or 80)}}$.
3110.	= $\frac{\text{Item 96} \times 1,000}{\text{Item 79 (or 80)}}$.
3110a.	= Item 3107 + Item 3108 + Item 3109 + Item 3110.
3111.	= $(1090.7 + 0.455 t_{10} - t_{12}) \left(\frac{(\text{Item 30} + 9 \times \text{Item 36})}{100} \right) + (\text{Item 49} - \text{Item 100}) \times \text{Item 51} \times 0.47 \times (t_{10} - t_4)$.
3112.	= Item 49 $\times (t_{10} - t_4) \times 0.24$.
3113.	= Item 3111 + Item 3112.
3114.	= $\frac{(\text{Item 98} - \text{Item 29}) \times 14,600}{100}$.
3115.	= $\frac{W \times (t_{14} - t_{13})}{\text{Item 79 (or 80)}}$. This item can be determined only in plants where ash and refuse are discharged into water.
3116.	= $\frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \text{Item 100} \times 10,160$ (use data from Item 43).
3117.	= $(\text{Item 45} - \text{Item 49}) \times (t_{10} - t_4) \times (0.24 + \text{Item 51} \times 0.47)$.
3118.	= $(t_7 - t_{10}) \times \left\{ \text{Item 49} \times 0.24 + \left[(\text{Item 49} - \text{Item 100}) \times \text{Item 51} + \frac{\text{Item 30} + 9 \times \text{Item 36}}{100} \right] \times 0.47 \right\}$.
3119.	= $(\text{Item 45} - \text{Item 49}) \times (t_7 - t_{10}) \times (0.24 + \text{Item 51} \times 0.47)$.
3120.	= $(\text{Item 46} - \text{Item 45}) \times (t_7 - t_i) \times (0.24 + \text{Item 51} \times 0.47)$.
3121.	= $(\text{Item 47} - \text{Item 46}) \times (t_7 - t_i) \times (0.24 + \text{Item 51} \times 0.47)$.
3122.	= Item 3106 - sum of Items 3107, 3108, 3109, 3110, 3111, 3112, and 3114 to 3121, inclusive.
3123.	= $(t_6 - t_{10}) \times \left\{ \text{Item 46} \times 0.24 + \left[(\text{Item 46} - \text{Item 100}) \times \text{Item 51} + \frac{\text{Item 30} + 9 \times \text{Item 36}}{100} \right] \times 0.47 \right\}$.

$$3124. = \frac{\text{Item 3107}}{\text{Item 3123}} \times 100.$$

TABLE S4B.—HEAT BALANCE OF STEAM-GENERATING UNIT COMPRISING BOILER, SUPERHEATER, REHEATER, ECONOMIZER, AND AIR HEATER. SOLID FUELS

<i>Heat Values</i>		
	B.t.u. Per Pound Fuel as Fired, Dry	Per Cent
4106. Heat per pound coal (as fired, dry) (higher heating value)		
4107. Heat absorbed by water in economizer.....		
4108. Heat absorbed by water and steam in boiler and super- heater.....		
4109. Heat absorbed by steam in superheater.....		
4110. Heat absorbed by steam in reheater.....		
4110a. Heat absorbed by steam-generating unit.....		
<i>Losses</i>		
4111. Heat loss due to combustible in refuse.....		
4112. Heat loss due to sensible heat in refuse.....		
4113. Heat loss due to incomplete combustion of carbon.....		
4114. Heat loss due to theoretical dry gases, t_2 to t_8		
4115. Heat loss due to moisture in coal, moisture accompany- ing theoretical air, and water from combustion of hydrogen, up to t_8		
4116. Heat loss due to excess air and accompanying moisture entering furnace, t_2 or t_8		
4117. Heat loss due to air and moisture leaking through boiler setting, t_1 to t_8		
4118. Heat loss due to air and moisture leaking through econ- omizer setting, t_1 to t_8		
4119. Heat loss due to air and moisture leaking through air- heater setting, t_1 to t_8		
4120. Heat loss due to unconsumed hydrogen and hydrocar- bons, radiation, and unaccounted for.....		
<i>Economizer</i>		
4121. Heat available to economizer in flue gases, including moisture, t_8 to t_{10}		
4122. Efficiency of economizer.....		
<i>Air Heater</i>		
4123. Heat available to air heater in flue gases, including moisture, t_7 to t_2		
4124. Heat absorbed by air heater.....		
4125. Efficiency of air heater.....		

NOTE.—If unit does not include economizer, do not fill out Items 4107, 4118, 4121, and 4122. If data are available for further segregation of losses, other items may be added.

Explanation of Computations for Table S4B

4106.	=	Item 31 (or 32).
4107.	=	$\frac{\text{Item 93} \times 1,000}{\text{Item 79 (or 80)}}$

- $$\begin{aligned}
 4108. &= \frac{\text{Item 94} \times 1,000}{\text{Item 79 (or 80)}}. \\
 4109. &= \frac{\text{Item 95} \times 1,000}{\text{Item 79 (or 80)}}. \\
 4110. &= \frac{\text{Item 96} \times 1,000}{\text{Item 79 (or 80)}}. \\
 4110a. &= \text{Item 4107} + \text{Item 4108} + \text{Item 4109} + \text{Item 4110}. \\
 4111. &= \frac{(\text{Item 98} - \text{Item 29}) \times 14,600}{100}. \\
 4112. &= \frac{W(t_{14} - t_{12})}{\text{Item 79 (or 80)}}. \text{ This item can be determined only in plants where ash and } \\
 &\text{refuse are discharged into water.} \\
 4113. &= \frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \text{Item 100} \times 10,160 \text{ (use data from Item 44)}. \\
 4114. &= \text{Item 49} \times (t_8 - t_2) \times 0.24. \\
 4115. &= (1090.7 + 0.455 t_8 - t_{12}) \times \frac{\text{Item 30} + 9 \times \text{Item 36}}{100} \\
 &\quad + (\text{Item 49} - \text{Item 100}) \times \text{Item 51} \times 0.47 \times (t_8 - t_2) \\
 4116. &= (\text{Item 45} - \text{Item 49}) \times (t_8 - t_2) \times (0.24 + \text{Item 51} \times 0.47). \\
 4117. &= (\text{Item 46} - \text{Item 45}) \times (t_8 - t_1) \times (0.24 + \text{Item 51} \times 0.47). \\
 4118. &= (\text{Item 47} - \text{Item 46}) \times (t_8 - t_1) \times (0.24 + \text{Item 51} \times 0.47). \\
 4119. &= (\text{Item 48} - \text{Item 47}) \times (t_8 - t_1) \times (0.24 + \text{Item 51} \times 0.47). \\
 4120. &= \text{Item 4106} - \text{the sum of Items 4107, 4108, 4109, 4110, inclusive, and 4111} \\
 &\text{to 4119, inclusive, and Item 4124.} \\
 4121. &= (t_6 - t_{10}) \times \left\{ \text{Item 46} \times 0.24 + \left[(\text{Item 46} - \text{Item 100}) \times \text{Item 51} \right. \right. \\
 &\quad \left. \left. + \frac{\text{Item 30} + 9 \times \text{Item 36}}{100} \right] \times 0.47 \right\}. \\
 4122. &= \frac{\text{Item 4107}}{\text{Item 4121}} \times 100. \\
 4123. &= \text{Item 50} \times 0.24 \times (t_7 - t_2). \\
 4124. &= (t_3 - t_2) \times \text{Item 50} \times (0.24 + \text{Item 51} \times 0.47). \\
 4125. &= \frac{\text{Item 4124}}{\text{Item 4123}} \times 100.
 \end{aligned}$$

TABLE L1A.—DATA AND RESULTS, TEST OF STATIONARY STEAM-GENERATING UNIT.
LIQUID FUELS—AS FIRED—MOISTURE FREE

General Information

Same as for boilers using solid fuels.

Descriptions, Dimensions, Etc.

11. Boiler heating surface.....	sq. ft.
12. Water-wall surface (water walls, screen or floor, and roof).....	sq. ft.
13. Superheating surface.....	sq. ft.
14. Economizer surface.....	sq. ft.
15. Reheater surface.....	sq. ft.
16. Total steam-generating unit surface.....	sq. ft.
17. Air-heater surface.....	sq. ft.
18. Number of burners.....	
19. Method of producing draft.....	
20. Fuel.....	
21. Area of furnace floor.....wide.....deep.....	sq. ft.

22. Height of furnace floor to nearest heating surface..... ft.
 23. Furnace volume per square foot of boiler heating surface..... cu. ft.

Fuel and Gas Analyses and Data

Fuel—As Fired—Moisture free

26. Moisture..... per cent
 27. Heating value per pound, as fired..... B.t.u.
 28. Heating value per pound, moisture free..... B.t.u.
 29. Flash point..... °F.
 30. Carbon..... per cent
 31. Hydrogen..... per cent
 32. Oxygen..... per cent
 33. Nitrogen..... per cent
 34. Baumé gravity..... deg.
 34a. Sulphur..... per cent

Gas

35. Gas analysis, furnace:

Per cent CO₂..... O₂..... CO..... N₂..... SO₂.....

36. Gas analysis, boiler outlet:

Per cent CO₂..... O₂..... CO..... N₂..... SO₂.....

37. Gas analysis, economizer outlet:

Per cent CO₂..... O₂..... CO..... N₂..... SO₂.....

38. Gas analysis, air-heater outlet:

Per cent CO₂..... O₂..... CO..... N₂..... SO₂.....

39. Dry gas per pound fuel in furnace (as fired, moisture free)..... lb.
 40. Dry gas per pound fuel at boiler outlet (as fired, moisture free)..... lb.
 41. Dry gas per pound fuel at economizer outlet (as fired, moisture free)..... lb.
 42. Dry gas per pound fuel at air-heater outlet (as fired, moisture free)..... lb.
 43. Dry gas per pound fuel, theoretical (as fired, moisture free)..... lb.
 44. Air supplied per pound fuel in furnace (as fired, moisture free)..... lb.

Pressures and Drafts

45. Moisture in air..... lb. per lb. air
 46. Steam pressure by gage, boiler..... lb. per sq. in.
 47. Steam pressure by gage, superheater outlet..... lb. per sq. in.
 48. Steam pressure by gage, reheater inlet..... lb. per sq. in.
 49. Steam pressure by gage, reheater outlet..... lb. per sq. in.
 50. Pressure of fuel at burners..... lb. per sq. in.
 51. Pressure of air for combustion at burners..... in. of water
 52. Draft in furnace..... in. of water
 53. Draft at boiler outlet..... in. of water
 54. Draft at economizer outlet..... in. of water
 55. Draft at air-heater outlet..... in. of water

Temperatures

56. Steam temperature at superheater outlet..... °F.
 57. Moisture in steam..... per cent
 58. Superheat..... °F.
 59. Moisture or superheat in steam entering reheater..... per cent or °F.
 60. Superheat in steam leaving reheater..... °F.
 61. Temperature of air surrounding boiler (*t*₁)..... °F.

62. Temperature of air entering air heater (t_2).....	°F.
63. Temperature of air leaving air heater (t_3).....	°F.
64. Temperature of air for combustion (t_4).....	°F.
65. Temperature of furnace (t_5).....	°F.
66. Temperature of gases leaving boiler (t_6).....	°F.
67. Temperature of gases leaving economizer (t_7).....	°F.
68. Temperature of gases leaving air heater (t_8).....	°F.
69. Temperature of feed water entering boiler (t_9).....	°F.
70. Temperature of feed water entering economizer (t_{10}).....	°F.
71. Temperature of water in boiler at point where gases leave boiler (t_{11}).....	°F.
72. Temperature of fuel at burner (t_{12}).....	°F.

Hourly Quantities

73. Duration of test.....	hr.
74. Fuel as fired per hour.....	lb.
75. Fuel per hour, moisture free.....	lb.
76. Fuel per cubic foot furnace volume per hour, as fired.....	lb.
77. Fuel per burner, per hour as fired.....	lb.
78. Fuel per burner, per hour, moisture free.....	lb.
79. Fuel per cubic foot furnace volume, moisture free.....	lb.
80. Actual water evaporated per hour.....	lb.
81. Steam through reheater per hour.....	lb.
82. Heat absorbed by water in economizer.....	B.t.u. per lb.
83. Heat absorbed by water and steam in boiler.....	B.t.u. per lb.
84. Heat absorbed by steam in superheater.....	B.t.u. per lb.
85. Heat absorbed by steam in reheater.....	B.t.u. per lb.
86. Rate of heat absorption in economizer.....	kB per hr.
87. Rate of heat absorption in boiler.....	kB per hr.
88. Rate of heat absorption in superheater.....	kB per hr.
89. Rate of heat absorption.....	kB per hr.
90. Total rate of heat absorption by steam-generating unit.....	kB per hr.

Evaporation

91. Rate of heat absorption per pound fuel (as fired).....	kB
92. Rate of heat absorption per pound fuel (moisture free).....	kB
93. Rate of heat absorption per square foot of steam-generating unit surface per hour.....	kB

Efficiency

94. Efficiency of steam-generating unit.....	per cent
95. Comparative efficiency of steam-generating unit (when not equipped with air heater).....	per cent

TABLE L1B.—HEAT BALANCE OF A STATIONARY STEAM-GENERATING UNIT. SHORT FORM. LIQUID FUELS

	B.t.u. per Pound Fuel as Fired, Moisture Free*	Per Cent
1095. Heating value of fuel.....		
1096. Heat absorbed by water in economizer.....		
1097. Heat absorbed by water and steam in boiler.....		
1098. Heat absorbed by steam in superheater.....		

1099. Heat absorbed by steam in reheater.....		
1099a. Heat absorbed by steam-generating unit.....		
1100. Heat loss due to moisture in fuel.....		
1101. Heat loss due to water from combustion of hydrogen...		
1102. Heat loss due to moisture in air.....		
1103. Heat loss due to dry chimney gases.....		
1104. Heat loss due to incomplete combustion of carbon.....		
1105. Heat loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted for.....		

* If test is on a moisture-free basis, draw a line through "as fired" wherever the words "as fired, moisture free" appear together, and *vice versa*. Whichever basis is chosen must be followed throughout. Percentage results in heat balance will be the same in either case.

Explanation of Computations for Tables L1A and L1B *Symbols*

Wherever CO_2 , O_2 , CO , and N_2 are used, they are the percentages by volume of these constituents in the gases of combustion.

h_g = total heat (in B.t.u. per pound) of saturated steam at the boiler-outlet pressure.

h_{g1} = total heat (in B.t.u. per pound) of superheated steam at the superheater-outlet pressure.

h_{g2} = total heat (B.t.u. per pound) of steam at reheater-inlet pressure.

h_{g3} = total heat (B.t.u. per pound) of superheated steam at reheater-outlet pressure.

h_{gw} = total heat (in B.t.u. per pound) in feed water at boiler inlet.

h_{gei} = total heat (in B.t.u. per pound) in feed water at economizer inlet.

h_{fg} = latent heat (in B.t.u. per pound) in steam at pressure in steam main.

t_s = temperature of steam after expansion in calorimeter.

kB = absorption of 1000 B.t.u.

11 and 12. **Boiler heating surface** shall consist of that portion of the surface of the **heat-transfer** apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system. This surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. Heating surface located in the furnace or furnace boundaries shall be measured as follows:

Tubes wholly exposed or wholly or partly embedded in refractory—that portion of the surface of the tubes which is exposed to the gas or refractory being cooled.

Tubes provided with extended surface—that portion of the surface of the tubes and the extensions which is exposed to the gas being cooled.

Tubes protected by blocks rigidly attached to the fire faces—that portion of the surface of the blocks which is exposed to the gas being cooled.

13. **Superheater surface** shall consist of that portion of the surface of the **heat-transfer** apparatus exposed on one side to the gas being cooled and on the other to steam being heated after leaving the boiler circulating system, the surface being measured on the flue-gas side.

14. **Economizer surface** shall consist of that portion of the surface of the **heat-transfer** apparatus exposed on one side to the gas being cooled and on the other to fluid being heated before entering the boiler circulating system, the surface being measured on the flue-gas side.

15. Reheater surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to steam being heated after expansion, the surface being measured on the flue-gas side.

16. The steam-generating unit heating surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas or refractory being cooled and on the other to the fluid being heated, measured on the side receiving heat. This item equals the sum of Items 11, 12, 13, 14, and 15.

17. Air-heater surface shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to air being heated for combustion, the surface being measured on the flue-gas side except that in air heaters of the regenerative type, where the surfaces are alternately in contact with gas and air, the air-heater surface shall be determined for the purpose of this code by multiplying the total surface of the apparatus by that proportion continuously in contact with the gas being cooled.

19. State whether natural, forced, induced, or a combination of one or more kinds of draft is employed.

20. State general class of fuel, give district where produced.

23. Furnace volume is the cubic space provided for the combustion of fuel before the products of combustion pass through any heating surface.

26 to 34. "As fired" analysis items = dry analysis items $\times \left(1 - \frac{\text{Item 26}}{100}\right)$.

39 to 42. $\frac{11\text{CO}_2 + 8\text{O}_2 + 7(\text{CO} + \text{N}_2)}{3(\text{CO}_2 + \text{CO})} \times \frac{\text{Item 30}}{100}$. See Eq. (76) (page 330).

43. = $\frac{12.52 \times \text{Item 30} + 26.56 \times \text{Item 31} + \text{Item 33} \times 1 + 5.325 \times \text{Item 34a}}{100}$.

44. = $\text{Item 39} + \frac{9 \times \text{Item 31}}{100} - 1$.

57. Determine either from charts or if a throttling calorimeter discharging into atmosphere (page 82) is used, by the following formula:

$$100 \times \frac{h_g - 1150.4 - 0.47 \times (t_s - 212)}{h_{fa}}$$

82. = $(h_{gw} - h_{ges})$.

83. = $(h_g - h_{gw})$.

84. = $(h_{g1} - h_g)$.

85. = $(h_{g3} - h_{g2})$.

86. = $\frac{\text{Item 80} \times \text{Item 82}}{1,000}$.

87. = $\frac{\text{Item 80} \times \text{Item 83}}{1,000}$.

88. = $\frac{\text{Item 80} \times \text{Item 84}}{1,000}$.

89. = $\frac{\text{Item 81} \times \text{Item 85}}{1,000}$.

90. = $\text{Item 86} + \text{Item 87} + \text{Item 88} + \text{Item 89}$.

91. = $\frac{\text{Item 90}}{\text{Item 74}}$.

92. = $\frac{\text{Item 90}}{\text{Item 75}}$.

93. = $\frac{\text{Item 90}}{\text{Item 16}}$.

94. = $\frac{\text{Item 91 (or 92)} \times 1,000}{\text{Item 27 (or 28)}} \times 100$.

$$95. = \frac{\text{Item 91} \times 1,000}{\text{Item 27} + (t_4 - 70) \times (\text{Item 44} \times 0.24)} \times 100 \text{ or}$$

$$= \frac{\text{Item 92} \times 1,000}{\text{Item 28} + (t_4 - 70) \times (\text{Item 44} \times 0.24)} \times 100.$$

Explanation of Computations for Table L1B (only)

$$1096. = \frac{\text{Item 86} \times 1,000}{\text{Item 74 (or 75)}}$$

$$1097. = \frac{\text{Item 87} \times 1,000}{\text{Item 74 (or 75)}}$$

$$1098. = \frac{\text{Item 88} \times 1,000}{\text{Item 74 (or 75)}}$$

$$1099. = \frac{\text{Item 89} \times 1,000}{\text{Item 74 (or 75)}}$$

$$1099a. = \text{Item 1096} + \text{Item 1097} + \text{Item 1098} + \text{Item 1099}.$$

$$1100. = \frac{\text{Item 26}}{100} \times (1090.7 + 0.455 t_6 - t_{12}).$$

$$1101. = \frac{\text{Item 31} \times 9^*}{100} \times (1,090.7 + 0.455 t_6 - t_{12}). \text{ If economizer is installed with-}$$

out air heater, substitute t_7 for t_6 . If air heater is installed, substitute t_8 for t_6 in Items 1100, 1101, 1102, and 1103.

$$1102. = \text{Item 44} \times \text{Item 45} \times 0.47 \times (t_6 - t_1). \text{ This item is small and is fre-}$$

quently included in Item 1103.

$$1103. = \text{Item 40 (or 41 or 42)} \times 0.24 \times (t_6 - t_1). \text{ If boiler and superheater, use}$$

Item 40; if economizer, use Item 41; if air heater, use Item 42.

$$1104. = \frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \frac{\text{Item 30} \times 10,160}{100}. \text{ Use data from Items 36, 37, or 38, de-}$$

pending on whether boiler and superheater are installed alone or with economizer but without air heater, or with air heater.

$$1105. = \text{Item 27 (or 28)} - \text{sum of Items 1096 to 1104, inclusive.}$$

* See footnote p. 318.

TABLE L2B.—HEAT BALANCE OF STEAM-GENERATING UNIT COMPRISING BOILER, SUPERHEATER, AND REHEATER WITH OR WITHOUT INTEGRAL ECONOMIZER.

LIQUID FUELS

Heat Values

	B.t.u. per Pound Fuel as Fired, Moisture Free*	Per Cent
2096. Heating value of fuel (higher heating value).....		
2097. Heat absorbed by water and steam in boiler.....		
2098. Heat absorbed by steam in superheater.....		
2099. Heat absorbed by steam in reheater.....		
2099a. Heat absorbed by steam-generating unit.....		

Unavoidable Losses

2100. Heat loss due to moisture in fuel, moisture accompany-
ing theoretical air, and water from combustion of
hydrogen, up to t_{11}

2101. Heat loss due to theoretical dry gases, t_4 to t_{11}		
2102. Total unavoidable losses.....		

Other Losses

2103. Heat loss due to incomplete combustion of carbon.....		
2104. Heat loss due to excess air entering furnace and moisture accompanying same, t_4 to t_{11}		
2105. Heat loss due to theoretical dry gases, moisture in fuel, moisture accompanying theoretical air, and water from combustion of hydrogen, t_{11} to t_6		
2106. Heat loss due to excess air entering furnace and moisture accompanying same, t_{11} to t_6		
2107. Heat loss due to air and moisture leaking through boiler setting, t_1 to t_6		
2108. Heat loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted for.....		

* See footnote, p. 315.

Explanation of Computations for Table L2B

2096.	=	Item 27 (or 28).
2097.	=	$\frac{\text{Item } 87 \times 1,000}{\text{Item } 74 \text{ (or } 75\text{)}}$
2098.	=	$\frac{\text{Item } 88 \times 1,000}{\text{Item } 74 \text{ (or } 75\text{)}}$
2099.	=	$\frac{\text{Item } 89 \times 1,000}{\text{Item } 74 \text{ (or } 75\text{)}}$
2099a.	=	Item 2097 + Item 2098 + Item 2099.
2100.	=	$(1,090.7 + 0.455 t_{11} - t_{12}) \times \frac{(\text{Item } 26 + 9 \times \text{Item } 31)}{100}$ + (Item 43 - Item 30) $\times$ Item 45 $\times$ 0.47 $\times$ ($t_{11} - t_4$).
2101.	=	Item 43 $\times$ ($t_{11} - t_4$) $\times$ 0.24.
2102.	=	Item 2100 + Item 2101.
2103.	=	$\frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \text{Item } 30 \times 10,160$ (use data from Item 36).
2104.	=	(Item 39 - Item 43) $\times$ ($t_{11} - t_4$) $\times$ (0.24 + Item 45 $\times$ 0.47).
2105.	=	$(t_6 - t_{11}) \times \left\{ \text{Item } 43 \times 0.24 + \left[\left(\text{Item } 43 - \frac{\text{Item } 30}{100} \right) \times \text{Item } 45 + \frac{\text{Item } 26 + 9 \times \text{Item } 31}{100} \right] \times 0.47 \right\}$.
2106.	=	(Item 39 - Item 43) $\times$ ($t_{11} - t_6$) $\times$ (0.24 + Item 45 $\times$ 0.47).
2107.	=	(Item 40 - Item 39) $\times$ ($t_6 - t_1$) $\times$ (0.24 + Item 45 $\times$ 0.47).
2108.	=	Item 2096 - sum of Items 2097, 2098, 2099, 2100, 2001 and 2103 to 2007 inclusive.

Other Boiler Test Codes.—For special forms of heat balance to be used with liquid fuels when the steam-generating unit is equipped with separate economizer and with air heater, and complete codes for gaseous fuel, see A.S.M.E. Test Codes "Stationary Steam Generating Units" (approved January, 1930, pp. 25-35).

COMPUTATION OF THE WEIGHT OF THE CHIMNEY GASES FROM THE ANALYSIS BY VOLUME OF THE DRY GAS

Two methods of calculating from the analysis by volume of the dry chimney gases the number of pounds of dry chimney gases per pound of carbon, or the weight of air supplied per pound of carbon, have been given by different writers. These may be expressed in the shape of formulas as follows:

$$(A) \text{ Pounds dry gas per pound C} = \frac{11\text{CO}_2 + 8\text{O}_2 + 7(\text{CO} + \text{N}_2)}{3(\text{CO}_2 + \text{CO})} \quad (76)$$

$$(B) \text{ Pounds air per pound C} = 5.8 \frac{2(\text{CO}_2 + \text{O}_2) + \text{CO}}{\text{CO}_2 + \text{CO}}. \quad (77)$$

in which CO_2 , O_2 , CO , and N are percentages by volume of the gases.

Formula (A) may be derived from the method of computation given in R. S. Hale's paper on "Flue-gas Analyses," *Transactions A.S.M.E.*, Vol. 18, page 902, and formula (B) from the method often given in treatises on steam boilers. Both are based on the principle that the density, relatively to hydrogen, of an elementary gas (oxygen and nitrogen) is proportional to its atomic weight, and that of a compound gas (carbon monoxide and carbon dioxide) to one-half its molecular weight. Both formulas are very nearly accurate when pure carbon is the fuel burned, but formula (B) is inaccurate when the fuel contains hydrogen, as the portion of the oxygen of the air supply which is required to burn the hydrogen is contained in the chimney gas as water vapor and does not appear in the analysis of the dry gas.

The following calculations of a supposed case of combustion of hydrogenous fuel illustrate the accuracy of formula (A) and the inaccuracy of formula (B). Assume that the coal has the following analysis: carbon, 66.50; hydrogen, 4.55; oxygen, 8.40; nitrogen, 1.00; water, 10.00; ash and sulphur, 9.55—total, 100 per cent. Assume that one-tenth of the carbon is burned to carbon monoxide, and nine-tenths to carbon dioxide; that the air supply is 20 per cent in excess of that required for this combustion; that the air contains 1 per cent by weight of moisture; and that the S in the coal may be considered as part of the ash. We then have the summary of results (shown on page 331) of the combustion of **100 pounds of coal**.

Professor D. S. Jacobus gives another formula for the air per pound of carbon, in which the error of Eq. (77) is almost entirely avoided. It is

Formula (C):

$$\text{Air per pound C} = \frac{7\text{N}}{3(\text{CO}_2 + \text{CO})} \div 0.77, \text{ or } \frac{\text{N}}{0.33(\text{CO}_2 + \text{CO})}, \quad (78)$$

in which N, CO₂, and CO are the percentages by volume of these gases. Making the computation from the data of the foregoing analysis, we have:

$$\text{Air per pound C} = \frac{80.656}{0.33(14.252 + 1.584)} = 15.434 \text{ pounds.}$$

The true value (as above) is 15.414 pounds.

	O from air	N = O × $\frac{11}{14}$	Total air	CO ₂	CO	H ₂ O
59.85 pounds C to CO ₂ × 2 $\frac{1}{2}$	159.60	534.31	693.91	219.45		
6.65 pounds C to CO × 1 $\frac{1}{2}$	8.87	29.70	38.57		15.52	
3.50 pounds H to H ₂ O × 8.....	28.00	93.74	121.74			31.50
	196.47	657.75	854.22			
1.05 pounds H to H ₂ O }						9.45
8.40 pounds H to H ₂ O }						
10.00 pounds water.....						10.00
1.00 pound N.....		1.00				
9.55 pounds ash and sulphur.....						
100.00						
Excess of air 20 per cent.....	39.29	131.55	170.84			
			1,025.06			
Moisture in air 1 per cent.....						10.25
Total weight gases, 1,125.76 pounds = ...	39.29	790.30		219.45	15.52	61.20
Total dry gases, 1,064.56 pounds.						
Per cent	O	N		CO ₂	CO	
Total dry gases, by weight.....	3.69	74.24		20.61	1.546	
Total dry gases, by volume.....	3.508	80.656		14.252	1.584	

Total gases = 1,125.76 + ash and sulphur (9.55) = 1,135.31 pounds total products.

Total air = 1,025.06 + moisture in air (10.25) + coal (100) = 1,135.31 pounds.

Dry gas per pound coal = 10.6456; per pound carbon = 10.6456 ÷ 0.665 = 16.008 pounds.

Dry air per pound coal = 10.2506; per pound carbon = 10.2506 ÷ 0.665 = 15.414 pounds.

Computation of the weight of dry gas and of air per pound carbon.

Formula (76):

$$\text{Dry gas per pound C} = \frac{14.252 \times 11 + 3.508 \times 8 + 82.240 \times 7}{3(14.252 + 1.584)} = 16.008 \text{ pounds.}$$

Formula (77):

$$\text{Air per pound C} = 5.8 \frac{2(14.252 + 3.508) + 1.584}{14.252 + 1.584} = 13.589 \text{ pounds.}$$

The error in the last result is 15.414 - 13.589 = 1.825 pounds.

CHAPTER XII

STEAM-ENGINE TESTING

Most important of the tests made of nearly all classes of machinery is the test for **mechanical efficiency**, which means the comparison of the useful work performed with the amount of work which it is theoretically possible to obtain with a perfect machine. In other words, in an engine the mechanical efficiency E_m is the ratio of the brake horsepower to the indicated horsepower, or

$$E_m = \frac{\text{b.hp.}}{\text{i.hp.}} \quad (79)$$

STEAM-ENGINE TESTING

TESTS FOR MECHANICAL EFFICIENCY AND FRICTION

Date..... Test made by.....
 Description of engine tested.....
 Tare of brake.....pounds Length of brake arm.....feet.....
 Engine and brake constants (see pp. 152 and 158).....

No. of read- ing	Time	Weight on brake, pounds		R.p.m.	Areas of indi- cator, cards, square inches		Indicated horsepower			Brake horse- power	Fric- tion horse- power	Mechanical efficiency, per cent
		Gross	Net		Head end	Crank end	Head end	Crank end	Total			

The difference between the indicated horsepower and the brake horsepower is called the **friction horsepower**. In many cases with very large engines, it is not readily possible to obtain the brake horsepower directly, and in such cases it is customary to obtain approximately the horsepower lost in friction from so-called "friction indicator diagrams," obtained from the areas of indicator diagrams when the only work done is that required to overcome its own friction or, in common parlance, when the engine is "running light." The brake horsepower is then taken to be the difference between the indicated horsepower and the

friction horsepower. Such a determination of friction horsepower and of mechanical efficiency by calculation cannot be considered very accurate, because the friction of an engine increases slightly with increasing loads.

Observed and calculated data of mechanical efficiency may be tabulated as shown in the table on page 332.

Valve Setting (Slide-valve Engines).—In order that steam may be used economically in an engine, it is necessary that the valve be set

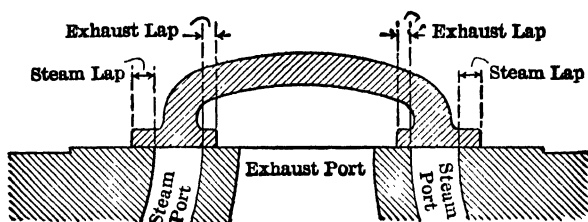


FIG. 257.—Ordinary D-slide valve in mid-position.

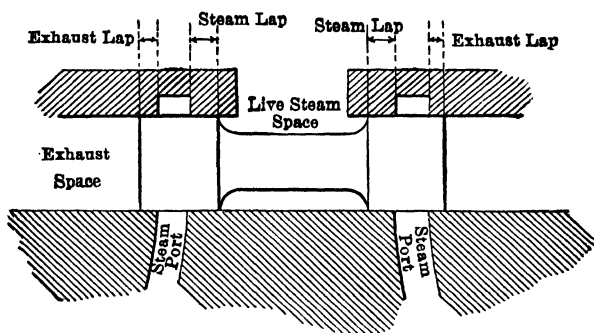


FIG. 258.—Piston type of slide valve in mid-position.

carefully and accurately, so that when an indicator card is taken the diagram obtained will be as nearly as possible like the ideal. Adjustment of a slide valve on an engine is accomplished in two different ways, with different effects:

1. By moving the valve on its stem.
2. By adjusting the eccentric.

Typical slide valves are shown in Figs. 257 and 258.

To Set the Valve for Equal Leads.—The first step in setting a valve is to place the engine on dead center and adjust the angle between the crank and eccentric so that the valve opens the port leading to the cylinder a slight amount. The width of the opening should be measured and recorded as a preliminary value of the lead on that end¹—assume, for example, it is $\frac{1}{8}$ inch. Then the engine should be placed on the opposite dead center and the port opening on that end measured and recorded as the preliminary value of the lead on that end—suppose it

¹ It is assumed, of course, that corresponding dimensions of the ports are the same at the two ends of the valve seat.

is $\frac{1}{8}$ inch. There is, then, a difference in lead on the two ends of $\frac{1}{8}$ inch. The valve must be moved on its stem a distance equal to half the difference, or $\frac{1}{16}$ inch. This movement of the valve will be in a direction **away from the port having the smaller opening.**

By the method described the two leads of the valve will be made the same; in other words, the distance the valve uncovers the steam ports when the engine is on the dead center will be the same at both ends of the cylinder. But while the leads are equal, they are not necessarily the required amount and it remains to set the eccentric to give the leads desired. Place the engine once more accurately on dead center and, after loosening the eccentric, move it on the shaft so as to change the lead to the amount desired. As a final check, after securing the eccentric, the engine can be placed on the other dead center to see that the lead is correct.¹

An engine can be put on dead center quite accurately by the **method of trammels.** When the engine is just a little off the center to be determined, make small scratch marks opposite each other both on the cross-head and on one of the guides. Now set a pair of dividers or trammels with one end resting on the bedplate of the engine, its foundation, or some convenient stationary object near the flywheel, and with the other end mark a point on the flywheel. The engine should then be moved over or beyond the dead center until the marks made on the cross-head and on the guide come together again. With the dividers set with the points the same distance apart as before, again put a mark on the flywheel. Then if the engine is turned back so that the end of the dividers used to mark on the flywheel is at a point halfway between the two marks, it will be set quite accurately on the dead center required. In all these adjustments care must be taken to turn the engine each time in the same direction with respect to the dead center so that the lost motion or backlash is taken up in the same direction. The engine must be placed accurately on center, because when the crank is near the dead center the eccentric is in such a position that a slight movement of the shaft causes considerable movement of the valve.

A somewhat similar method can be used to find the dead center of an engine with an enclosed crank case, like a multicylinder automobile engine. In an engine of this kind, there are no cross-heads, each connecting rod being attached directly to a wrist pin in the engine piston, as in Fig. 259. In the application of the "trammel" method to this case, two narrow marks are made opposite each other, one on the flywheel and the other on the engine frame as at *C*, when the piston is in some position off dead center like that in the figure. Through a spark-plug

¹ This method applies explicitly only to a valve like the one in Fig. 257, which takes steam on the "outside." When the valve takes steam on the inside (Fig. 258), the eccentric must be moved in the opposite direction.

hole (or priming cock hole, if there is one) the distance between the top of the piston as at *A* and the outside surface of the cylinder at *B* should be accurately measured. Obviously, the top of the piston should be clean (not covered with a layer of carbon) if an accurate determination is to be made. Then the engine is turned over its *upper* dead center till the distance between *A* and *B* is the same as before. In this position make a mark on the engine frame opposite the new position of the mark *C* that was originally made on the flywheel. Assume for clearness in making the diagram that this mark on the engine frame is at *D*. Then the engine will be on dead center when the mark *C* on the flywheel is at *E* which is midway between *C* and *D*.

To set the valve for equal cut-offs, the valve is first set on its stem to make its movement, for convenience, approximately symmetrical with the ports. Now an adjustment of the eccentric must be made so that steam will be cut off at each end of the cylinder when the piston has moved the same distances or the same percentage of the stroke from the two ends. To perform this adjustment, mark on the cross-head as accurately as possible the limits of the stroke, and set the cross-head at the percentage of the stroke for cut-off appearing to be most suitable for the conditions of load. Move the eccentric on the shaft in the direction in which the engine is to run until it can be seen that the valve would be just closing the steam port at the end of the cylinder from which the piston is moving. Fasten the eccentric securely in this position and turn the engine over to observe whether the valve will be just closing the other steam port when the piston has moved the same distance, measured on the cross-head, from the other end of the cylinder. If the setting is not correct, the error should be halved, correcting for one-half the error by moving the valve on the stem and for the other half by moving the eccentric on the shaft. This operation, which is a "cut-and-try" process, must be repeated until the required setting is secured.

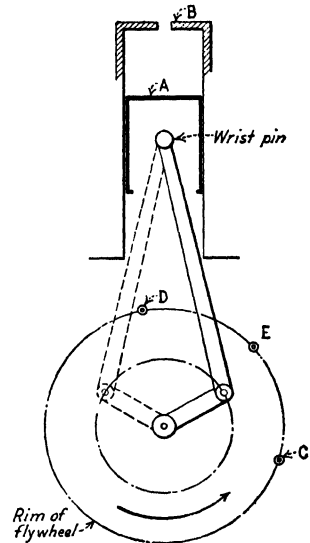


FIG. 259.—Dead-center method for engine with enclosed crank case.

Methods similar to those described are preferred usually for the accurate setting of the slide valves of slow- and medium-speed engines. High-speed engines as well as slow-speed engines of the Corliss type have their valves set usually on the basis of the information secured from indicator diagrams taken on the engines, showing approximately the "timing" of the events of the stroke. To set a **slide valve** successfully

by the "indicator" method, the valve and ports should be measured to determine the "lap" dimensions and port openings indicated in Fig. 257 (page 333), as well as the valve travel. With these data a **Zeuner valve diagram** should be constructed, showing a good steam distribution for assumed lead or cut-off. Then construct the theoretical indicator card from the Zeuner diagram and adjust the setting of the valve on the stem and the eccentric on the shaft until a close approximation to the theoretical card is obtained. In this adjustment the first thing to be

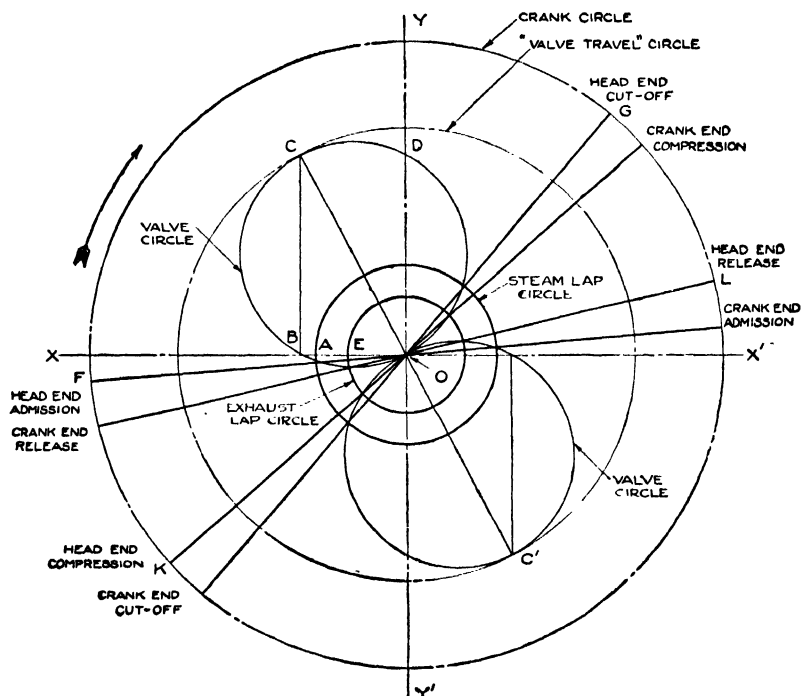


FIG. 260.—Zeuner valve diagram.

done is to equalize the travel of the valve by locating it on its stem so that the travel will be the same on both sides of its mid-position.

Valve setting is often expedited by referring during the process of setting to valve diagrams, constructed by either the Zeuner or the Bilgram method. Although the Bilgram method is excellent for designers, it is not so convenient as the Zeuner for valve-setting requirements. The two methods are explained briefly in the following paragraphs:

Zeuner Diagram.—Draw horizontal and vertical intersecting lines XX' and YY' as in Fig. 260. At O as a center draw two circles, one having a diameter equal to¹ the stroke of the engine and the other having

¹ In making valve diagrams "equal to" dimensions may, of course, be taken to convenient scales.

a diameter equal to the valve travel. From O lay off the distance equal to the steam lap OA plus the lead AB . At B erect a perpendicular and from the intersection of the perpendicular with the "valve travel" circle draw the line COC' . The angle YOC is the angle of advance. With CO and $C'O$ as diameters draw two "valve circles." When the crank has revolved to the position OY , the valve has traveled from its mid-position a distance equal to OD . Valve distances may be obtained similarly for other crank positions. With O as a center draw two circles, one having a radius equal to the steam lap and the other having a radius equal to the

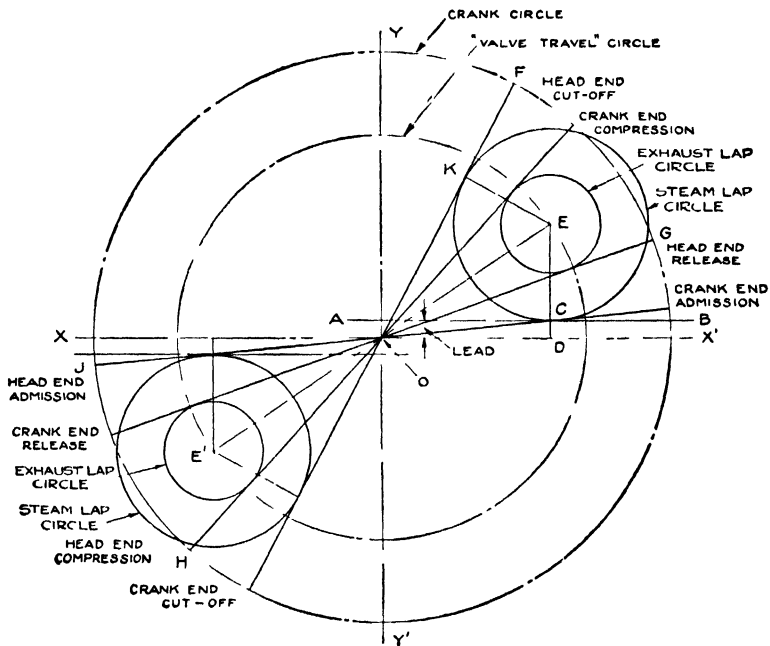


FIG 261 Bilgram valve diagram

exhaust lap, as, for example, OA and OE , respectively. Through the intersection of the “valve circles” $OBCDO$ and the “steam lap circle” draw the lines OF and OG . The line OF represents the position of the crank when the *admission* at the head end is taking place as the valve has moved a distance from the mid-position equal to the steam lap OA . By the same reasoning the line OG represents the crank position when *cut-off* occurs at the head end. Through the points where the “valve circles” and “exhaust lap circle” intersect draw the lines OK and OL . The line OL represents the crank position when release occurs and OK the crank position when compression occurs at the head end. By the same method the crank positions for the events at the crank end may be determined.

Bilgram Diagram.—Draw horizontal and vertical lines XX' and YY' as in Fig. 261. From their intersection O as a center draw a circle having

a diameter equal to¹ the stroke of the engine and another circle having a diameter equal to the valve travel. Lay off AB away from XX' at a distance equal to the lead CD and parallel to XX' . With a radius EC equal to the steam lap draw a circle having its center on the "valve travel circle" and tangent to CD . Having located the point E draw the line EOE' . The angle EOD is the angle of advance. The displacement of the valve from its mid-position for any crank position can be obtained by dropping a perpendicular from E to the line representing the crank position. For example, when the crank is at OF the amount of travel of the valve from its mid-position is KE . The distance KE is equal, however, to the steam lap, so that the cut-off occurs at the head end when the crank is at OF . Likewise other events may be obtained. OJ is the crank position for admission, OG for release, and OH for compression. By similar reasoning the position of the crank for the crank-end events can be found.

When checking valve setting by the indicator method, use a spring in the indicator to give a diagram about $1\frac{1}{2}$ inches high so that events of the stroke will be shown as clearly as possible. Sometimes it is difficult to determine these events on a diagram on account of the curves gradually running into each other without the point separating the different curves being clearly defined. A good method for such cases is to produce along their regular trend both of the curves of which the intersection is required and take for the intersection the point where these curves cross each other. The method is illustrated on an indicator card in Fig. 267 (page 361) showing the point of cut-off.

In a slide-valve engine it is not possible to set the valve to secure at the same time equal cut-offs and equal leads.

The following table shows the effect on the events in the stroke of an engine of changing the angle of advance and the length of the valve stem:

	Effect of increasing			
	Angle of advance		Stem length	
	Head end	Crank end	Head end	Crank end
Outside lap.....	Same	Same	Increases	Decreases
Inside lap.....	Same	Same	Decreases	Increases
Lead.....	Increases	Increases	Decreases	Increases
Admission.....	Earlier	Earlier	Later	Earlier
Cut-off.....	Earlier	Earlier	Earlier	Later
Release.....	Earlier	Earlier	Earlier	Later
Compression.....	Earlier	Earlier	Later	Earlier

¹ In making valve diagrams "equal to" dimensions may, of course, be taken to convenient scales.

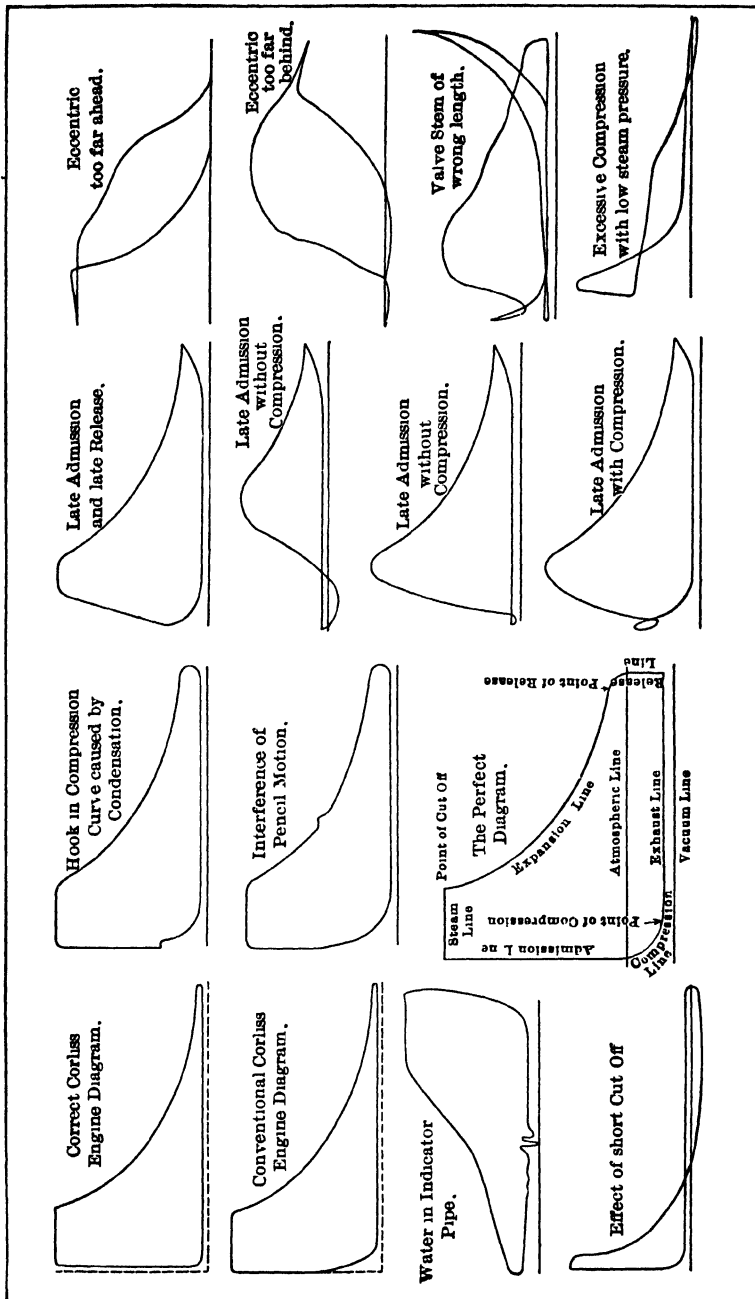


Fig 262—Samples of ideal and imperfect indicator diagrams (From Power.)

Ideal and imperfect indicator diagrams taken from slide-valve and Corliss engines are shown in Fig. 262. A little study of such diagrams may help to solve many difficulties in valve setting.

Setting Corliss Valves.—A brief description¹ of the essential parts of the valve gear of a Corliss engine will assist in obtaining a clearer conception of the subject. In Figs. 263 and 264 similar letters of reference indicate the same parts of the mechanism.

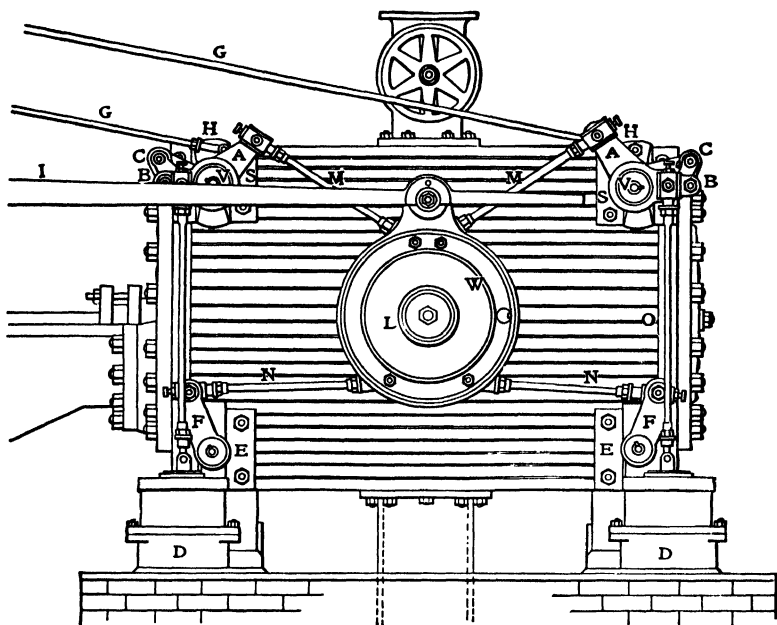


FIG. 263.—Corliss valve gear.

Figure 263 shows all the essential parts of the valve gear. The steam valves work in the chambers *S, S* and the exhaust valves work in the chambers *E, E*. The double-armed levers *AC, AC* work loosely on the hubs of the valve-stem brackets and the lever arms *B, B*; the former are connected to the wrist plate *W* by the rods *M, M*; the levers *B, B* are keyed to the valve stems *V, V* and are also connected by the rods *O, O* to the dashpots *D, D*. The double-armed levers carry at their outer ends *C, C* hardened steel catch plates, which engage with arms *B, B*, making the two arms *B* and *C* work in unison until steam is to be cut off; at this point another set of levers *H, H*, connected by the cam rods *G, G* to the governor, come into play, causing the catch plates to release the arms *B, B*, the outer ends of which are then pulled downward by the weight of the dashpot plunger, causing the steam valves to rotate on their

¹ This description is mainly from *Am. Machinist*, Vol. 18, p. 391. For clearness the article is considered unusually good.

axes and thus cut off steam. These are the essential features of the Corliss gear, although the design of the mechanism is greatly modified by different builders.

The exhaust-valve arms *F* are connected to the wrist plate by the rods *N*, *N*, and it is seen that all the valves receive their motion from the wrist plate; the latter receives its motion from the hook rod *I*; this rod is generally attached to a rocker arm, not shown; to this arm the eccentric rod is also attached. The carrier arm is usually placed about midway between the wrist plate and eccentric and in the center of its travel stands in a vertical position.

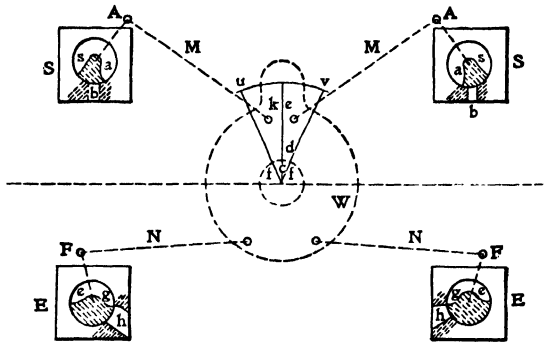


FIG. 264.—Diagram of a Corliss valve mechanism.

The setting of the valves is not a difficult matter when, on the wrist plate, its support, valves and cylinder, the customary marks have been placed for finding the relative positions of wrist plate and valves.

Now in referring to Fig. 264, when the back bonnets of the valve chambers have been taken off, there will be found a mark or line *a* on the end of each valve *s*, *s*, coinciding with the working or opening edges of each valve; another line *b* will be found on each face of the steam-valve chamber coinciding with the working edge of the valve, and the line *h*, on the face of each exhaust-valve chamber, coincides with the working edge of the exhaust port. On the hub of the wrist plate will be found a line *d*, coinciding with the center line *d*, *k*; lastly, there are three lines *f*, *c*, *f*, on the hub of the wrist-plate support, placed in such a way that, when the line *d* coincides with the line *c*, the wrist plate will stand exactly in the center of its motion and, when the line *d* coincides with either of the lines *f*, *f*, the wrist plate will be at one of the extreme ends *u* or *v* of its travel. It should be noticed that, since the lines *f*, *e*, *f* are drawn on the periphery of the hub of the wrist-plate support, and the line *d* is drawn on the periphery of the wrist-plate hub, these lines cannot stand in a vertical line, as shown. This way of showing them has been adopted simply for the purpose of making the matter plain.

In setting the valves, the first step will be to set the wrist plate in its central position, so that lines *c* and *d* will coincide, and fasten the wrist plate in this position by placing a piece of paper between it and the washer *L* on its supporting pin. Now set the steam valves so that they will have a slight amount of lap, that is to say, the lines *a*, *a* must have moved a little beyond the lines *b*, *b*. The amount of this lap depends much on individual preference and experience. It ranges from $\frac{1}{16}$ to $\frac{1}{4}$ inch for small engines, and from $\frac{1}{4}$ to $\frac{9}{16}$ inch for comparatively large engines. This lap is obtained by lengthening or shortening the rods *M*, *M* by means of the adjusting nuts.

Now by lengthening or shortening the rods *N*, *N* and by moving the adjusting nuts, place the exhaust valves *e*, *e* in a position so that the working edges will just open the exhaust ports, or, in other words, place the lines *g* and *h* nearly in line with each other. Some engineers prefer a slight amount of lap, others prefer a slight opening of the exhaust ports when the valves are in this position; under these conditions the lines *g* and *h* cannot be in line but will stand apart, as indicated in the diagram. The distance between these lines will, of course, be equal to the desired amount of opening. For small engines it is about $\frac{1}{16}$ inch, and for larger engines may be increased to $\frac{8}{16}$ inch, but in any case the amount of this opening should be less than the lap of the steam valves, otherwise there will be danger of steam blowing through without doing work.

The paper between the wrist plate and the washer on the supporting pin should now be taken out, so that the wrist plate connected to the valves can be swung on its pin.

The next step will be to give some attention to the rocker arm. Set this arm in a vertical position by means of a plumb line, and connect the eccentric rod to it, then turn the eccentric around on the shaft, and see that the extreme points of travel are at equal distances from the plumb line. To secure this a little adjustment in the stub end of the eccentric rod may be necessary. Now connect the hook rod *I* to its pin on the wrist plate, and again turn the eccentric around on the shaft, and thus determine the extreme points of travel of the wrist plate. If all parts have been correctly adjusted, the line *d* will coincide with the lines *f*, *f* at the extreme points of travel; if this is not the case, the hook rod will have to be adjusted at its stub end so as to obtain the desired equalized motion of the wrist plate.

The next step will be to set the valves correctly with respect to the position of the crank. To do so the lengths of the rods *M*, *M*, *N*, and *N* must not be changed, but the following procedure should be followed: Place the crank on one of its dead centers, and turn the eccentric loosely on the shaft in the direction in which the engine is to run, until the steam valve nearest to the piston shows an opening or lead of $\frac{1}{8}$ to $\frac{1}{4}$ inch,

according to size of engine, the smaller lead, of course, being adopted for small engines. After the proper lead has been given to this valve, secure the eccentric, and turn the shaft with eccentric in the same direction in which the engine is to run until the crank is on the opposite dead center, and notice if the opening or lead at this end of the cylinder is the same as on the other steam valve; if not, shorten or lengthen slightly, as may be necessary, the connection between wrist plate and eccentric. Much adjustment in the length of these connections is not permissible without resetting the valves with reference to the wrist plate.

The only thing which now remains to be done is to adjust the cam rods *G, G*. To do this, secure the governor balls in their highest position, and disconnect the hook rod from wrist pin. Lengthen or shorten the cam rods *G, G*, so as to bring the detachment apparatus into action, swing the wrist plate back and forward, and make such adjustments in the rods *G, G* as to permit the steam valves to be released when the steam port has been opened about $\frac{1}{8}$ inch. This adjustment is for the purpose of keeping the engine under the control of the governor, in case, for some reason or another, the load on the engine is suddenly thrown off. After this adjustment the governor balls should be placed in their lowest position, in which the releasing gear should not detach the steam valves but allow the steam to enter nearly full stroke. Sometimes the releasing gear is constructed in such a manner as to close the steam valves automatically, in case the belt leading to the governor should be broken, or the load on the engine suddenly thrown off. In cases of this kind the governor balls need not be placed in their highest position but should be placed in their lowest position, and the wrist plate moved to either end of its extreme travel. The steam port opposite this end of travel of wrist plate will then be wide open. Now adjust the corresponding cam rod so that the releasing gear is nearly on the point of releasing the valve; then move the wrist plate to the other end of its extreme travel and adjust the other cam rod in the same manner. To prove the correctness of the cut-off adjustment, raise the governor balls to about the position they would be when at work, or to a medium height, and block them there; then, with the connection made between the eccentric and the wrist plate, turn the engine shaft slowly in the direction in which it is to run, and when the valve is released measure upon the slide the distance through which the cross-head has moved from its extreme position. Continue to turn the shaft in the same direction, and when the other valve is released measure the distance through which the cross-head has moved from its extreme position, and if the cut-off is equalized, these two distances will be equal to each other. If they are not, adjust the length of the cam rods until the points of cut-off are at equal distances from the beginning of the stroke. Replace the back bonnets and see that all connections have been properly

made, which will complete the setting of the valves. Wherever convenient, it is desirable that an indicator be applied to the engine when at work, and the setting of the valves tested. If necessary, they should be readjusted for the best possible condition for economical work.

Clearance Determination of an Engine.—This test is made usually to determine the clearance volume of a steam or a gas engine. It is sometimes important to know the clearance volume of an engine, as it materially affects the expansion curve of the engine. If it is too large, it causes an excessive loss in the engine. The clearance volume is also necessary if a theoretical expansion curve is to be constructed.

The engine is first set on dead center with the piston at the head end of the cylinder. This is done easily by the "method of trammels" (see page 334). Then the steam-chest cover and valve are to be removed and a rubber gasket under a block of wood is placed over both steam-port openings in the valve seat and bolted on. Usually candle wicking must be packed around the piston to stop excessive leakage. For this purpose the cylinder head must be removed and again replaced.

A.S.M.E. Method of Clearance Determination.—To carry out this method, set the engine exactly on the center, with the piston at the end of the cylinder where the clearance is to be determined. From a quantity of water previously weighed pour enough into the clearance space by means of a funnel through some available opening, such as may be obtained by removing a steam valve in a horizontal four-valve cylinder, or through the holes for indicator cocks, until the clearance space is completely filled. By weighing the water remaining, the amount necessary to fill the clearance space can be obtained, and by making the proper allowance for temperature the clearance volume may be calculated.

Care should be taken to see whether the clearance space is so arranged that no air is retained when it is filled with water. If such pockets cannot be properly vented, this method will not give accurate results. When full, the gradual lowering of the water will show leakage by the valves or piston. Allow leakage to go on for a certain time, then from a quantity of water previously weighed pour enough into the clearance space to fill it up again, note the time it takes to do this, and weigh the water remaining after filling up the clearance space. Then the weight of water from which clearance space is to be calculated (W_1) can be determined as follows:

$$W_1 = W - \frac{wT}{t + t_1},$$

where W = weight of water filled into clearance space, first time.

T = time to fill clearance space.

t = time allowed for leakage.

w = weight of water for filling space up again.

t_1 = time required to fill clearance space again.

This method will give the proper correction to be subtracted if the leakage is not very great. If the leakage is too great, and the object of the test will allow it, the valves and the piston rings must be put in proper condition.

All water must, of course, be drained from each end before filling. Removing the cylinder head for packing the piston is the best method for observing with certainty that there is no water in the head end. The drip pipes in the cylinder can usually be relied on to remove water from the crank end. Determinations should be repeated several times, and the average value is to be used in calculations.

The percentage of clearance is then found by dividing the clearance volume thus found in cubic inches by the volume displaced by the piston in one stroke, also in cubic inches.

The area of **interior steam surfaces** is all of that part of the interior of the steam cylinder, the piston head and the parts back to the valves that are exposed to the steam at the time that the exhaust valve is opened. If desired, the **clearance surfaces** can be calculated by subtracting from this interior steam surface the area of the circumferential walls of the cylinder from the location of the piston at the dead-end center to the location of the piston when the exhaust valve first opens.

Instrument for Measuring Engine Clearance Volumes.¹—An instrument for measuring engine clearance volumes has been developed at the U. S. Bureau of Standards. This instrument is built so that it can be used with any engine having a clearance volume up to 48 cubic inches. Figure 265 shows the general arrangement of the instrument when connected to an engine cylinder to be tested.

The principles upon which the instrument is based are as follows: The pressure in the cylinders *a* and *b* is increased by means of a weight or a force acting upon two cylindrical copper bellows *A*. The volumes *A*, *a*, and *b* will be decreased by the same amount by the movement of the bellows. Because of the change in volume there will be a pressure increase in each cylinder, the greater increase of pressure occurring in the smaller volume. This difference of pressure will be indicated by the difference in liquid level in the U-tube connecting the two cylinders. In practice, the clearance volume *a* is unknown, while the volume *b* can be changed by the movement of the piston *c*. The volume of cylinder *b* must be calibrated by comparison. By changing this volume (increasing

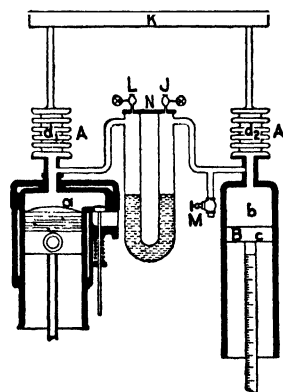


FIG. 265.—Apparatus for measuring cylinder volume.

¹ See *Tech. Notes 27*, National Advisory Committee for Aeronautics, Automotive Power Plants Section, U. S. Bur. Standards.

or decreasing), the pressure in the cylinder can be made the same, as shown by the liquid in the U-tube remaining level. The volumes a and b are then equal. This is only true provided there is no leakage or if the rate of leakage for the two cylinders is the same. Owing to an engine cylinder never being strictly tight, a means for permitting an equal rate of leakage from the comparison cylinder must be provided. This adjustment is made by setting the valve M which is connected to the pipe line between the cylinder b and the U-tube, so that the rates of leakage for the two cylinders are the same.

The actual instrument has a vernier and graduated scale on the piston stem which permits direct reading of volumes up to 48 cubic inches in steps of $\frac{1}{16}$ cubic inch. In case larger volumes are to be measured, an auxiliary cylinder can be connected to the pipe line near cylinder b . The auxiliary volumes d_1 and d_2 are equal and therefore do not affect the operation of the instrument so far as the measurement of the clearance volume is concerned.

Testing Surface Condenser Leakage.¹—Whenever a steam-rate test of an engine or turbine is to be conducted by weighing or measuring the condensed steam (condensate) discharged by a surface condenser, the amount of leakage of circulating water into the steam space of the condenser shall be determined and, if it exceeds the value specified in the Test Codes for Steam Turbines or Reciprocating Steam Engines, shall be reduced to such value in order to establish satisfactory test conditions for the engine or turbine.²

Some of the methods of conducting condenser leakage tests can be used with salt circulating water only, and others may be used with either fresh or salt circulating water. Some require accurate apparatus and an experienced personnel and can be used only when these are available. Some of the methods are carried on when the engine or turbine is running for the regular steam-rate tests. It is desirable to use such methods when possible because the condenser leakage may be greater when the engine or turbine is running than when it is not, owing to temperature changes and vibration. Some of the methods are to be used when the

¹ The amount of water or steam used in sealing glands, valves, and joints of the engine or turbine, hot-well pump, and other auxiliaries is not condenser leakage. Any external water used for such purposes must be accounted for independently of condenser leakage, during a steam-rate test or a condenser-leakage test. It is highly desirable to use condensed steam (condensate) taken from the line between the hot-well pump and the weighing tank for all sealing purposes. If this is not done and if water from some external source is mixed with the condensed steam, there is the additional complication of keeping track of such water, as well as an error, due to the different condition of such water, when the electrolytic or silver nitrate method is used for determining the amount of condenser leakage.

² Abridged from "Code on Instruments and Apparatus," Part XXI, Power Test Codes of A.S.M.E., 1928.

engine or turbine is not running, immediately before and after the steam-rate tests. Low leakage should be shown with such methods because of the possibility of increase when the engine or turbine is running.

Qualitative Silver Nitrate Method.—This method is suitable only for circulating water containing an appreciable amount of salt, such as sea water or a tidewater mixture of sea and fresh water. It can be used with fresh water, provided it is mixed with brine or otherwise chlorinated, as, for example, at the suction of the circulating pump. It is the preferred method for such waters when the condenser is thought to be very tight. The test is to be made when the engine or turbine is running. The method depends upon the chemical reaction of silver nitrate with chlorides which are abundant in salt water. Samples of condensate are tested by means of a silver nitrate solution in order to determine if a trace of salt is present. If no trace is found, the condenser is known to be perfectly tight. If a trace is found, the quantitative silver nitrate method, as described below, must be used.

Quantitative Silver Nitrate Method.—This method is suitable for use with salt circulating water. It is to be used when it is not certain that the condenser is tight enough for the qualitative method and is the preferred method for such a case. It shall also be used when the qualitative method has been used and has shown some leakage. It is to be made when the engine or turbine is running. Samples of pure condensed steam which has not passed through the condenser, of condensate from the condenser, and of circulating water are titrated against a silver nitrate solution (using potassium chromate as an indicator) in order to determine the concentration of chlorides in each. Then by calculation the amount of circulating water leaking into the steam space of the condenser may be determined. This method requires some readily obtained chemical apparatus and some easily acquired skill. Complete instructions are given on page 349, so that with a little effort the test may be carried out accurately.

Short Direct-weight Method.—This method is used with fresh circulating water only and is used only when the engine or turbine is not running. It consists in shutting off all steam supply to the engine or turbine running the circulating pumps at the same speed as during the steam-rate test, maintaining as nearly as possible the same vacuum in the steam space as during the steam-rate test, and weighing the amount of circulating water that leaks into the steam space as it is removed from the hot well of the condenser.

Standard Direct-weight Method.—This method is the same as the *short direct-weight method*, with the addition of some refinements which make for greater precision.¹ It is the *preferred method* with fresh circulating water.

Electrolytic-conductance Method.—This method involves a determination of the relative resistance offered to the flow of electric current between electrodes immersed in samples of pure condensed steam from the steam space of the condenser, of condensate from the hot well of the condenser, and of circulating water. From these data the amount of circulating water leaking into the steam space may be calculated. The test is to be made when the engine or turbine is running. It requires a rather elaborate apparatus and an experienced personnel. The greater the amount of solids in solution in the circulating water, the easier it is to use this method.

With fresh circulating water, if the amount of solids in solution is appreciable and constant, and if the apparatus and personnel are appropriate, this method is preferable to the short direct-weight method. Extreme care must be exercised if the circulating water contains less than 20 grains of dissolved solids per gallon.

¹ The precision details are explained on p. 354.

Cold-water Pressure Method of Locating Condenser Leaks.—A common method is to open or remove the outer heads of the condenser and to fill the entire steam space of the condenser with water. To obtain a pressure sufficient to show the small leaks which must be eliminated, there should be a head of approximately 20 feet above the center line of the condenser. In order to have such a head, the turbine must be filled with water. All shaft seals must be packed with putty or packing of some kind, and condenser relief valves, automatic valves, etc., must be fastened down. **This method, while often used, is very difficult and is not recommended.**

Vacuum Method of Locating Condenser Leaks.—Open or remove the inlet and outlet water boxes.¹ Plug one end of all of the condenser tubes with tapered, softwood plugs, pushed in by hand. Then start the air pump so as to obtain at least 18 inches vacuum. It may be necessary, in order to secure this amount of vacuum, to seal the shafts of the air pump with putty, or it is also possible to seal the shafts with steam seals. However, the admission of steam will often make it difficult to maintain the desired amount of vacuum. When a satisfactory vacuum is obtained, a small gas flame or a lighted candle is to be applied to the open end of each of the tubes and to the entire tube sheet. Even a small leak will be indicated when the flame is pulled into the tube, tube sheet, or packing. The more usual types of leaks are as follows: cracked tube, corroded tube, fine holes in tube, and leaky packing.

Collection of the samples of condensate, circulating water, and pure condensed steam shall be simultaneous or nearly so. This is especially important if the circulating water is tidewater with variable salt content. The samples shall be collected in clean flasks or rubber-stoppered bottles. The pipe from which a sample is collected shall be thoroughly cleaned by a preliminary flow; the bottle or flask shall then be thoroughly rinsed with the same water as that with which it is to be filled, and finally the sample shall be collected. The conduits from which the sampling pipes are led shall be main-line pipes carrying the flow and not dead-end pipes carrying motionless fluid. Samples shall not be less than 1 liter. Samples of condensate shall be obtained from a cock in the condensate-pump discharge pipe or from the flow into the measuring tanks. When the condensate is sampled during the conduct of a steam-rate test, if the quantity of condensate drawn off in rinsing and sampling is appreciable, it shall be caught in a bucket and weighed, and the weight added to the weight of condensate used in computing the steam rate. Samples of circulating water shall be collected from a cock at any convenient point. Of course, a point must be selected where the pressure is above that of the atmosphere, and not in a siphon or suction line, or else a vacuum connection must be attached to the sampling vessel. Samples of pure condensed steam shall always be collected in a way which gives a representative sample of the steam entering the condenser unaffected by condenser-leakage or other contaminating factors. The sample of condensed steam may be obtained from the main steam line just preceding the turbine, or preferably from a turbine stage—one with pressure just above that of the atmosphere. Such a sample may be collected by a long drip pipe cooled with wet rags or by a black tin or glass coil immersed in an ice bath and connected up with rubber tubing. There also may be used an air-cooled condenser of about 20 feet or more of pipe, arranged in any convenient form of coil or bank. This coil shall be thoroughly blown out with steam before the beginning of the test. Often it is desirable to have the coil tinned on the inside. However, tin melts at 450°F. and cannot be used with high-temperature steam.

With such an air condenser, a continuous flow of condensed steam shall be maintained during the test. The condensed steam should be collected in an annealed glass or other flask which will not break due to the high temperature. However, the sample of condensed steam always shall be at room temperature when being used.

¹ The **water boxes** of a surface condenser are the outside compartments to which the pipes for circulating water are attached.

When the condensed steam is collected from the steam main and in cases where it is impossible to avoid the objectionable practice where water is injected into the steam to maintain a given temperature, or if the steam contains a comparatively large percentage of salt, the condensed steam shall be taken from the center of the main in such a manner that the drops of water condensing on the side of the main and containing solids, cannot enter the drip pipe. This is for the reason that salt is much more readily carried by water than steam.

Instead of collecting the condensed steam from the steam main or from a turbine stage, it may be collected from the interior of the condenser, as is required for the electrolytic conductance method.

Where the condenser has been leaking and has been repaired or where it has not been in use for a period exceeding 12 hours, *samples shall be considered as preliminary* until about 2 hours have elapsed from the time the turbine is at test load, because the condenser collects solids while idle, and the first samples indicate a leakage greater than the actual. Sometimes it may be desirable to run the condenser for short periods at different loads so that various steam currents will wash all parts of the condenser. The dropping of the leakage to a reasonably constant or a negligible value will indicate when the regular test may be started.

Regular samples shall be taken at convenient intervals, at least every 30 minutes, during the entire progress of the regular steam-rate tests. Deductions from the steam flow for condenser leakage shall be made on the basis of the average of all of the values for a test point shown by the regular samples.

Details of Qualitative Silver Nitrate Method.—This method may be used with salt or brackish circulating water. A silver nitrate solution is prepared by dissolving about 1 gram of chemically pure silver nitrate, AgNO_3 , in about 2,000 cubic centimeters (2 liters) of distilled water. Exact measurement is not required.

If the salt content of the circulating water ever becomes negligible, the silver nitrate method cannot be used. If there is any doubt of this, the circulating water shall be tested for salt by mixing 1 cubic centimeter of circulating water with about 100 cubic centimeters of condensate and using the method for testing the condensate described in the next two paragraphs. The silver nitrate test may or may not be used, according as there are or are not perceptible white clouds.

A few drops of the silver nitrate solution shall be dropped into each sample of condensate. The slightest trace of salt will be shown by a white cloud which will follow each drop of silver nitrate solution as it falls in the sample. If such clouds occur, or if the final mixture is whitish, there is leakage, and either the steam-rate test shall be discontinued and the condenser repaired, or the quantitative method shall be started at once and continued throughout the steam-rate test. A log sheet shall be kept even when no leakage is shown.

This test is very sensitive and, if there are no white clouds or perceptible traces of white in the final mixture, it may be certain there is no leakage. A tight condenser will give such a negative result. Sometimes there may be a slight trace of blueness due to negligible traces of leakage, which may be disregarded. Of course, if there is any doubt as to whether there is evidence of leakage, it must be assumed that there is leakage, and the quantitative test started.

Details of Quantitative Silver Nitrate Method.—This method may be used with salt circulating water or with fresh circulating water mixed with brine or otherwise chlorinated.

Apparatus required for the test comprises a *burette* graduated to tenths of a cubic centimeter and with a total capacity of at least 25 cubic centimeters, with *stand* (a in Fig. 266), *filling funnel*, etc.; (b) *beaker*; and (c) *Erlenmeyer flask* for the sample being titrated or else a 150 cubic centimeter white porcelain *casserole*; (d) 100 cubic centimeter *cylinder* graduated to 1 cubic centimeter; (e) small graduated cylinder with

which 10 cubic centimeters can be measured accurately, or (f) 10 cubic centimeter pipette; (g) glass-stoppered bottles for silver nitrate solution, and for potassium chromate indicator, or (h) indicator bottle (the silver nitrate bottle should be of brown or amber glass); (i) three rubber-stoppered bottles or flasks for samples being tested; (j) a 1-liter annealed-glass flask to collect the condensed steam; (k) a 1-gallon bottle of distilled water; (l) about $\frac{1}{2}$ liter silver nitrate solution, AgNO_3 ; and (m) about 20 cubic centimeters of potassium chromate solution, K_2CrO_4 . Each piece of apparatus must be *chemically clean*, and each must be rinsed with the sample with which it is to be filled before *each* use.

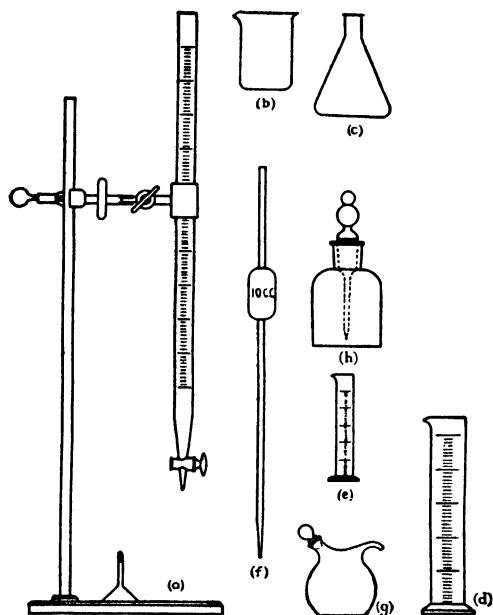


FIG. 266.—Apparatus for quantitative silver nitrate method.

Water used for rinsing and diluting in this work shall be pure distilled water. This distilled water may be obtained from an outside source or, if it is properly salt-free, from the source of the condensed steam.

If the silver nitrate used exceeds 5 per cent of the amount necessary for a sample of the circulating water as prepared for test, the water tested is not sufficiently salt-free to be used for rinsing and diluting in this work and shall be replaced by a better supply.

The volume of standard silver nitrate solution required for 20 tests (one day) is about $\frac{1}{2}$ liter. The volume of potassium chromate solution required for the same tests is about 20 cubic centimeters. The solutions used should be as dilute as is consistent with a sharp end point in the titration. It may be noted that the strength of solution does not affect the accuracy of the results when each set of samples is titrated with solutions of the same strength. It will be found most convenient, however, to use solutions of a uniform strength throughout. A solution containing 4.8 grams of chemically pure silver nitrate per liter of distilled water will be taken as standard. One cubic centimeter of this solution is equivalent to 1 milligram of chlorine, to 1.66 milligrams of sodium chloride, and to about 1.85 milligrams of total salts in sea water. As an indicator, a 10 per cent solution of chemically pure potassium chromate is satisfac-

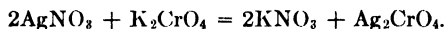
tory. This will be in the proportion of 60 grams of K_2CrO_4 to 1,000 grams of distilled water.

At various stages of the test, as explained later, *samples must be titrated*, and in each case the following method is to be used and must be conducted by a man who has had some practice. The titration method is as follows: (a) Rinse the 100 cubic centimeter graduate and Erlenmeyer flask thoroughly with pure distilled water. Measure out 100 cubic centimeters (unless otherwise specified) of the sample to be tested in the 100 cubic centimeter graduate. The sample must be at room temperature as heat affects the titration. Pour this 100 cubic centimeter sample into the Erlenmeyer flask or casserole. (b) Rinse the burette with distilled water and arrange it on the stand as shown in Fig. 266. Fill the burette with silver nitrate solution to the zero graduation (the zero is at the top). (c) Place a few drops (exactly the same number in every titration) of the potassium chromate indicator into the flask containing 100 cubic centimeters of the sample to be tested. This will give the solution a light-orange color. Whirl the solution in the flask a bit to make the color uniform. Note the reading of the burette on the log sheet and then place the flask under the burette. Allow the silver nitrate solution to drip into the sample to be tested. The spout shall be in the center of the mouth so that the silver nitrate will not run down the side of the flask. When the drops begin to turn red, the flow of silver nitrate shall be stopped and the sample thoroughly shaken. Unless too much silver nitrate has been added, the red color in parts of the sample will disappear when the flask is shaken. Silver nitrate shall then be added a few drops at a time until a permanent red color remains. The base of the stand on which the flask rests should be clear white to assist the observation of color, or a white background should be secured in some similar way if a porcelain dish has not been used. This permanent red color gives the *end point of the titration* and the reading of the burette shall then be noted on the log sheet. The difference between the two readings will be the volume of silver nitrate required for the sample being tested.

The reaction between silver nitrate and sodium chloride



yields white compounds which do not affect the color of the sample, which is a light orange, due to the potassium chromate indicator. As soon as all of the sodium chloride (together with other chlorides which may be present) is consumed, however, a very slight excess of silver nitrate reacts with the potassium chromate to form silver chromate, a dark-red compound:



The end point of the titration is clearly indicated by the sharp change of color.

If there is a possibility that sodium hydroxide or sodium sulphate, or both, have been carried over from the boiler, the solution to be titrated must previously be made neutral as described below, to avoid white clouds due to silver hydrate or silver sulphate.

Add two drops of phenolphthalein indicator. If the solution colors red, add a very dilute solution of sulphuric acid ($N/50$) until the red color just disappears and then three drops in excess. If, upon the addition of the phenolphthalein, the solution does not color, add a very dilute solution of sodium carbonate ($N/50$) until the solution becomes slightly red, then add the dilute acid until the color disappears, and then three drops in excess. By this procedure the chloride determination will be made in a neutral solution and none of the silver nitrate will be used up in forming silver hydrate or silver sulphate.

After the preliminaries already given are arranged, the test is made as follows, with each set of samples. The results are recorded as in the sample log sheet (page 352).

The solutions from the same silver nitrate bottle shall be used for each set of samples: First titrate 100 cubic centimeters of the condensate sample. The 100 cubic centimeter sample of condensed steam is titrated next in the same manner as the condensate. If this requires the same amount of reagent as the condensate, it is not necessary to test the sample of circulating water, as the equality of salt content indicates no addition of salt from the circulating water and hence no condenser leakage.

If, however, the circulating water is to be tested it shall be diluted before titration. This is done by mixing together, in the 100 cubic centimeter graduated cylinder, 10 cubic centimeters of the circulating water sample, and 90 cubic centimeters of the *salt-free* distilled water. These are mixed by inverting the cylinder several times after plugging it with a clean rubber stopper. From the thoroughly mixed dilute sample, 10 cubic centimeters are measured and mixed with 90 cubic centimeters of distilled water. This 1 per cent solution of circulating water is titrated in exactly the same manner as the sample of 100 cubic centimeters of condensate.

In terms of the following symbols: C_h = volume of silver nitrate required for 100 cubic centimeters of condensate, C_w = volume of silver nitrate required for the 100 cubic centimeters of the 1 per cent solution of circulating water; C_s = volume of silver nitrate required for the 100 cubic centimeter sample of condensed steam. Then the percentage of circulating water mixed with the condensed steam, that is, the **percentage leakage** L , is

$$L = \frac{C_h - C_s}{C_w - \frac{C_s}{100}} \text{ per cent}$$

or very closely

$$L = \frac{C_h - C_s}{C_w} \text{ per cent.}$$

For example, if $C_h = 2$ cubic centimeters, $C_s = 0.4$ cubic centimeter, and $C_w = 16$ cubic centimeters,

$$L = \frac{2.0 - 0.4}{16} = \frac{1.6}{16} = 0.1 \text{ per cent leakage.}$$

Observations and results should be recorded as in the following sample log sheet.

Sample Log Sheet.

CONDENSER LEAKAGE—QUANTITATIVE SILVER NITRATE TEST

Time: Dec. 22, A.M.	Condensate, cubic centi- meters			Condensed steam, cubic centimeters			Circulating water, cubic centimeters			Per cent leakage
	Burette reading		Amount of reagent consumed net	Burette reading		Amount of reagent consumed net	Burette reading		Amount of reagent consumed net	
	Before	After		C _s	Before		After	C _{st}		Before
8:30	18.3	19.8	1.5	20.0	20.3	0.3	5.1	22.1	17.0	0.07
9:00	20.1	21.4	1.3	20.5	20.7	0.2	6.3	22.8	16.5	0.07
9:30	18.7	19.7	1.0	20.0	20.4	0.4	6.7	23.0	16.3	0.04
10:00	23.4	24.3	0.9	23.0	23.4	0.4	2.3	19.7	17.4	0.03
10:30	20.2	20.9	0.7	19.7	20.1	0.4	5.8	24.3	18.5	0.02
11:00	2.5	3.1	0.6	24.3	24.7	0.4	3.7	21.1	17.4	0.01

The run made after the steam-rate test, may show 50 per cent greater leakage than specified in the Test Code for Steam Turbines (page 381) or Reciprocating Steam Engines (page 356). If there is greater leakage than this after the steam-rate tests, they shall be discarded, and repeated after the condenser has been repaired.

The tests shall be made immediately before and after the regular steam-rate test. If any appreciable time is allowed to elapse, the condition of the condenser may change. If there is a series of steam-rate tests, as, for instance, when tests are made at different loads, the condenser-leakage tests may be made before the first and after the last one. However, it is also advisable to have intermediate condenser-leakage tests in such a case.

Preparation for Short Direct-weight Test.—All possibility of entrance of steam, water, or condensate into the condenser during the condenser-leakage test shall be eliminated.

Usually the turbine or engine, with its steam piping, is connected directly to the condenser. In such event each steam inlet to the turbine or engine must be disconnected or shown to admit no steam by use of two valves with a drip or bleeder between them. During the leakage test there is usually vacuum on the turbine and, with leaky valves, steam or water might be sucked in and give apparent condenser leakage. This may be avoided by the use of a large bleeder having a diameter about 10 per cent of the steam-pipe diameter, or by making certain that there is no suction on the bleeder by feeling it or noting if a tube connected to the bleeder will suck water from a vessel.

If the turbine has openings for *steam extraction* or *bleeding* for heating feed water or for operation with low-pressure steam, or, in fact, openings of any kind, the above-mentioned precautions must be taken or else this method of test shall not be used. No steam or water shall be used on the seals of shaft glands, valves, or joints. Openings usually so sealed may be closed with rubber tubing or putty. All other openings through which air might enter the condenser must, of course, be carefully sealed. The vacuum must be as nearly as possible the same during the condenser-leakage test as during the steam-rate test. If there is a valve between engine or turbine and the condenser, it shall be closed in order to minimize air leakage.

During the condenser-leakage test there may be vacuum on parts not normally under vacuum, so that there is more chance for air leakage. Painting the outside of the engine or turbine, with shellac or asphaltum with vacuum on, is often necessary in order to secure good vacuum.

A **dry air pump** is understood to mean a pump which takes nothing but air and saturated steam from the condenser, whether or not water is used in the pump itself.

A **wet air pump** is understood to mean a pump which takes both water and air from the condenser. During these tests it must be impossible for water to enter a wet air pump from any source except the condenser itself.

During this test the circulating and air pumps are operated. The circulating pump shall be run at the same rate as during the regular steam-rate tests. An air pump, wet or dry, shall be operated at such a rate as will give a vacuum as nearly as possible the same as during the steam-rate tests. During the test, some circulating water may leak into the steam space, all of which shall be collected and weighed¹ in the regular tanks in the same manner as specified for the flow during the steam-rate tests. Observations shall be taken every 10 minutes of the condenser vacuum. Throughout the test, by any method, the temperature of the condenser shell must be lower than the boiling point corresponding to the condenser absolute pressure, in order to prevent loss by evaporation. From the time and the weight of the condenser leakage, the leakage per hour shall be found. The average of the values of

¹ Detailed instructions for the collection of condenser leakage are given in A.S.M.E. Power Test Code, Part XXI, Chap. I, p. 11.

condenser leakage before and after the steam-rate tests gives the leakage per hour, which must be subtracted from the flow per hour measured during the steam-rate test.

Preparation for Standard Direct-weight Method.—When this method is used, all of the directions for the Short Test shall be followed with the following additional details. This method may be used only with fresh circulating water.

When the condenser-leakage test is ready to be started, the air pump shall be run for a short time and an observation made of the vacuum which will occur during the leakage test. If the vacuum is not within 1 inch of mercury of the value expected during the regular steam-rate tests, additional efforts shall be made, by better sealing, attention to air pump, etc., to secure a better vacuum. After this has been done, if the vacuum is not within 2 inches of mercury of the value during the steam-rate test (page 356), and if it is not positively known that the pressure of the circulating water, during the leakage test is within 2 inches of mercury of the pressure during the steam-rate test, gages shall be used to give the circulating-water pressure during both the condenser-leakage test and the steam-rate tests. These shall be two mercury columns or gages of any approved kind connected to the water space of the condenser near the circulating-water inlet and outlet, respectively. They shall give either the direct pressures of the circulating water or the differential pressures between the condenser vacuum and the circulating water inlet and outlet, respectively. If direct gages are used, the differential pressures shall be computed, careful account being taken of the pressures above and below atmospheric pressure. If there is a poor vacuum, and if the siphon effect on the circulating water is great, it may happen that the absolute pressure in the steam space is greater than the absolute pressure at the water outlet, which gives a differential pressure in the opposite sense from that occurring during the steam-rate test, so that the direction of leakage is reversed and the leakage test is absurd.

When the leakage test is to be made by the standard method, where there is a **dry air pump**, a run is to be made for a preliminary period of 10 to 30 minutes before the regular condenser-leakage run, in order to drain the condenser, establish steady conditions, fill water pockets, etc. The circulating pump is, of course, to be kept running. The procedure is as follows: (a) close hot-well drain and vent; (b) start air pump; (c) run for 10 to 30 minutes; (d) stop air pump; (e) open drain and vent, and let leakage run off. After the first flow there will be a trickle from the drain, which may gradually decrease for a long period or may settle down to a small steady flow. The rate of flow should be found by collecting it for 5 to 10 minutes in a small vessel. If the rate of flow is visibly decreasing, it should be measured successively until it becomes constant or becomes less than the allowable limits.

During the **regular leakage run**, readings shall be taken every 10 minutes showing the condenser vacuum and the differential pressure between the steam space and the water space near inlet and outlet. Corrected value of condenser leakage may be calculated in terms of the following symbols:

p_{r1} and p_{r2} = observed values in any convenient units of differential pressures between steam space and water spaces at condenser inlet and outlet during steam-rate tests.

p_{l1} and p_{l2} = the same, during condenser-leakage test, using the same units as above.

L = corrected value of condenser leakage in pounds per hour.

L_0 = observed condenser leakage in pounds per hour obtained by dividing the actual leakage in pounds (taking account of tare) by the number of hours of the leakage run.

The leakage is proportional to the square root of the differential pressure, and the average leakage may therefore reasonably be assumed to be proportional to the average of the square roots of the differential pressures at inlet and outlet. It follows that

$$L = L_0 \times \frac{\sqrt{p_{r1}} + \sqrt{p_{r2}}}{\sqrt{p_{l1}} + \sqrt{p_{l2}}}$$

The values of L from the leakage tests before and after the steam-rate tests must be seen to be allowable within the limits. The average of the values of L from leakage tests before and after the steam-rate test gives the leakage per hour, which must be subtracted from the flow per hour measured during the steam-rate test.

Electrolytic-conductance Method.—The usual system of applying the electrolytic-conductance method to the measurement of condenser leakage involves a determination, by a proper electrical instrument, of the conductance, or resistance to flow of electric current, between electrodes immersed in samples of condensate, pure condensed steam, and circulating water. For this purpose a number of methods are available, any one of which, when properly used, will give satisfactory results. Good commercial instruments of several types are on the market, and one of these may be used, such as the Dionic water tester, distributed by James G. Biddle, the Leeds and Northrup conductivity bridge, or the Esterline-Angus concentration meter. The detailed instructions of the manufacturer are, of course, to be followed, as well as the general instructions herein. An apparatus may also be especially prepared, according to instructions given by Caldwell in *Power*, Aug. 9, 1921.¹

A.S.M.E. TEST CODE (1926) FOR RECIPROCATING STEAM ENGINES (ABRIDGED)

The Code for steam-engine tests applies to tests for **determining the performance of the engine alone** (including reheaters and jackets, if any), apart from that of the independently driven auxiliaries which are necessary for its operation, and apart from that of feed-water heaters or other apparatus for reclaiming heat. For tests of an engine and independently driven auxiliaries combined with means for reclaiming heat (for example, a multiple-expansion engine from the receiver of which steam is withdrawn for heating feed water or other purposes), the necessary changes in test procedure are readily made, or the test outline given in the Test Code for Complete Steam Power Plants² should be followed.

Measurements.—The measurements will consist of some or all of the following quantities: (a) cylinder diameters and stroke; (b) volumetric clearance in percentage of the piston displacement; (c) diameters of piston rods and tail rods; (d) indicated horsepower; (e) brake horsepower or shaft horsepower output; (f) kilowatt output if engine is connected to a generator; (g) speed in revolutions per minute; (h) pressure in the steam pipe before the throttle; (i) barometric pressure; (j) percentage of moisture or number of degrees of superheat in the steam just before the throttle; (k) back pressure in the exhaust pipe near the engine cylinder; (l) pressures in receivers; (m) temperature of the exhaust steam leaving all cylinders; (n) temperature of the drips if any, from jackets, reheaters, and receivers; (o) temperature of steam leaving receivers, cylinder jackets, reheaters, and similar parts; (p) weight of return drips from jackets, reheaters, and receivers; (q) weight of the condensed steam in pounds or the

¹ Detailed methods of procedure and calculations will be found in *Mech. Eng.*, Vol. 47, No. 11, p. 925.

² See p. 402. This test code for complete steam power plants has not been officially adopted by the Power Test Committee of the A.S.M.E.

weight of the water fed to boilers, less drips and leakage if the test is based on feed-water measurement; (r) temperature of the condensed steam; (s) engine-room temperature and outside-air temperature; (t) variation in steam pressure in the steam chest.

Instruments and apparatus required for a performance test of a reciprocating steam engine are:

(a) Tanks and platform scales for weighing water (or water meters calibrated in place); (b) graduated scales attached to the water glasses of the boilers if the feed water is measured; (c) pressure gages, vacuum gages, or mercury columns and thermometers provided with suitable mercury wells; (d) steam calorimeters; (e) barometer; (f) steam-engine indicators; (g) planimeter; (h) tachometer, revolution counter, or other speed-measuring apparatus; (i) friction brake or dynamometer, if available; (j) appropriate electrical instruments if engine is connected to an electric generator.

Dimensions.—The dimensions of engine cylinders should be taken when they are cold. If extreme accuracy is desired as in scientific investigations, corrections should be applied to the cold dimensions to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. The **clearance of the cylinders** may be determined approximately from working drawings of the engine. For accurate work, when practicable, the clearance should be determined by the **water-measurement method** (page 344).

Steam Rate.—The steam rate or steam consumption of an engine equipped with a surface condenser should be determined by weighing the condensate from the condenser, assurance being obtained that all the steam entering the cylinders passes into the condenser. When water is weighed, scales shall be fitted with tanks arranged so that a continuous flow can be cared for, and the scales shall not show an error greater than 2 in 1,000.

When water is measured, the volumetric tanks used may be of the types described on page 220. They shall be accurately calibrated by actual weighing, and a correction shall be determined and applied for water temperature.

Precautions in Condenser-leakage Tests.—In any test in which the total steam is determined by weighing or measuring the condensate from a surface condenser, the condenser shall be tested for leakage of cooling water into the condensed steam. If this leakage exceeds the limits given below, the condenser is not in condition suitable for a reliable test, and it shall be made satisfactorily tight before the test. These limits depend on the rated capacity of the engine and are expressed as a percentage of the rated, or guaranteed, steam flow at rated load.

	Per Cent
Less than 700 horsepower, or 500 kilowatts.....	1.00
700 to 1,400 horsepower, or 500 to 1,000 kilowatts.....	0.75
Over 1,400 horsepower, or 1,000 kilowatts.....	0.50

If a condenser leakage is determined by silver nitrate titration (for salt water, page 347) or by the electrolytic-conductance method (page 355), observations shall be made every half hour during the engine steam-rate test, in addition to the observations made during the preliminary leakage test. If the observations made during the steam-rate test indicate an increase of condenser leakage above the allowable limit, the steam-rate test shall be discontinued, and the condenser shall be made satisfactorily tight before resuming the engine test.

The water level in the condenser hot-well or inlet pipe to the condensate pump shall be checked at the beginning and end of a run, and at intermediate points which mark the division of the run into periods, for each of which the steam rate is to be determined; and correction shall be made for any outstanding change in the quantity of water stored therein. It is highly desirable that means be employed for holding the condensate level constant throughout the test, as, for example, by throttling the discharge of the condensate pump. One satisfactory way of obtaining the desired result is to note that a gage glass on the inlet pipe to the condensate pump shows a constant level. If the volume of the pipe between the gage glass and the condensate pump is small, it will be sufficient to see that the gage glass is always empty. The resulting error can easily be computed and will usually be negligible. It shall be determined that there are no other points at which water can accumulate. If such places cannot be avoided, proper correction shall be made for each of them just as for the hot well. All lines leading from and to the condenser, other than the air and condensate pump suction and turbine drips, shall be blanked off or fitted with two valves with a bleeder between them or, if this is impossible, means shall be provided for determining the quantity of water passing through each such line. If a surface condenser is not available, the steam rate should be determined by feed-water tests, which require the measurement of the various supplies of water fed to the boiler, of the water wasted by separators and drips on the main steam line, of the steam used for other purposes than the main engine cylinders, and of water and steam which escape by leakage from the boiler and piping. All of these last quantities must be determined and be deducted from the total measured feed water supplied to the boiler. The heat consumption is determined from the steam rate. **The uncertainties of making these measurements are such that steam rates determined by this method can be considered as only approximate.**

With suitable precautions, the *weighing or measuring of boiler feed water* should give results accurate to within plus or minus 3 per cent. To secure even this degree of accuracy, it is necessary to reduce the effect of the inevitable uncertainties of this method by the exercise of the greatest care and by the prolongation of each constant-load run to not less than 5 hours' duration and 10 hours if possible.

In making an engine test where the steam consumption is determined from the amount of water discharged from a surface condenser, leakage of the piston rods and valve rods, etc., should be guarded against; for if this is excessive, the test is of little use, as the leakage consists partly of steam that has already done work in the cylinder and partly of water condensed from the steam in contact with the cylinder.

The flow of steam to a reciprocating engine is of a **pulsating** character. The steam meters so far developed will not serve, therefore, to measure the quantity of steam. If the flow does not **fluctuate**, however, it is possible by use of the carbon dioxide method described in the October, 1916, issue of *Mechanical Engineering* to measure a pulsating flow within 1 or 2 per cent.

The steam consumed by independently driven auxiliaries which are required for the operation of the engines should not be included in the total steam from which the steam consumption is calculated. These should be supplied from a separate boiler, if possible, if the steam consumption is determined by the measurement of feed water.

Starting and Stopping.—The engine and appurtenances should first be thoroughly heated and run under the prescribed conditions until uniformity is secured. If the period to be covered by the test includes the time employed in warming up a cold engine, as in the practice of many industrial plants, the preliminary heating should be omitted.

When a surface condenser is used, the test should start by commencing to weigh or measure the condensate and any other quantities of steam consumption involved and at the same time beginning the regular observations and other necessary test work. At the end of the allotted time the test is stopped by discontinuing the measurements and observations. When feed-water measurements are employed, the test should be started by carefully observing the steam pressure and water level in the steam drums of the boiler and the level in the feed tank, if measuring tanks are used, and at the same time beginning the water measurements and taking up the regular work of the test. At the end of the prescribed time the water levels and steam pressure should be brought as near as practicable to the same points as at the start, and the observations discontinued. If there are differences in the water levels or pressure, proper corrections must be applied to the water measurements.

To guard against the loss of all the data obtained, should an accident compel the stopping of the test before the time planned, it will be found advisable at certain periods throughout the test to observe simultaneously the steam pressure, the water level in the boiler, the level in the feed tank, and other conditions which must at the end of the test agree with the starting conditions.

When feed-water measurements are employed, care should be taken, where the activity of combustion affects the height of water, that the

same conditions of fire and draft are secured at the end as at the beginning of the test. Care should also be observed to note the average height of water in the glass when the water line fluctuates.

When surface-condenser measurement is employed and the load is substantially constant, the test should be continued for such time as may be necessary to obtain a number (not less than four) of successive 15-minute records during which the results are reasonably uniform.

Records.—Instruments should be read and indicator cards taken from each end of each cylinder at least quarter hourly when the conditions are uniform and oftener when there is much variation. By permitting the pencil to trace over the diagram several times, a pencil band will result, thus giving a better average than a single line for power measurement. If there are wide fluctuations in readings, they should be shown by recording instruments. Each indicator card should be marked with the number, date, time, scale of spring and end of cylinder, and on one card of each set the readings of the steam gages should be recorded. The log should contain the record of the readings of steam and vacuum gages, thermometers, calorimeters, speed indicator, load-measuring devices, and all other instruments, and these readings should be obtained at practically the same time the indicator diagrams are made. The areas, lengths, mean effective pressures, and cut-offs shown by the indicator diagrams should also be entered in the log. If complete test data are desired, representative steam-pipe diagrams should be taken with an indicator applied near the throttle-valve gage and operated by connection to the reducing motion of the cylinder indicators.

A set of specimen indicator cards should be carefully selected from the whole number taken, and these should be embodied in the record. The specimen cards selected should be such as to show the average conditions of pressure and cut-off. If steam-pipe diagrams are obtained, specimens of these should also be placed in the record.

Throttle Pressure.—The throttle pressure, or the average pressure in the steam pipe just before the throttle, is that shown by a corrected steam gage attached to the steam pipe one and one-half to two diameters from the throttle. Fluctuations in this pressure may be reduced by a moderate choking of the gage cock. When greater precision is desired or when the fluctuations in pressure are large, the average pressure may be found by working up the steam-pipe indicator diagrams taken at or near the same point and finding the mean pressure for the entire stroke, during the periods of admission.

Steam Consumption.—Whether the steam supply is wet, dry, or superheated, the actual steam consumption is stated in the report of the test. When the engine is supplied with wet steam, the quantity of dry and saturated steam comprised in the wet steam is found by deducting from the total weight of steam as measured the moisture as shown by a

calorimeter near the throttle. If the calorimeter indicates moisture in excess of 2 per cent, the steam-rate test cannot be regarded as reliable. Superheated steam requires no correction.

The steam consumption corrected to correspond to any desired or specified set of conditions as to moisture, superheat, pressure, and vacuum which differ from the conditions of the test should be determined as explained in the Test Code for Steam Turbines (page 380).

In view of the fact that "dry steam" is practically unobtainable commercially, and that the presence of moisture is detrimental to the economy of the engine to a greater degree than the amount present in the steam would indicate, appropriate corrections, referred to above, for correcting the actual steam rate for quality are to be used in arriving at a value of "corrected steam" (page 362).

Heat Rate B.t.u. Consumption.—The heat rate of an engine is found by multiplying the steam rate in pounds of steam per horsepower-hour, by the difference between the heat content of 1 pound of steam, in B.t.u., at the average pressure and of the average quality found in the steam pipe near the throttle, and the heat content of 1 pound of water, in B.t.u., at the temperature of saturated steam at the average pressure existing in the exhaust pipe near the cylinder.

Thermal Efficiency.—The proportion of the total heat consumption which is converted into work is called the "thermal efficiency" and is found by dividing 2,545 (B.t.u. equivalent to 1 horsepower-hour¹) by the heat rate measured in B.t.u. per horsepower-hour. The quotient is multiplied by 100 to express the thermal efficiency in percentage.

The formula is:

$$\text{Thermal efficiency} = \frac{2,545}{w(h_{g1} - h_{g2})},$$

where w = pounds of steam as supplied per indicated horsepower-hour.

h_{g1} = total heat per pound of steam at the initial conditions prevailing before throttle valve, in B.t.u. per pound from steam tables.

h_{f2} = heat per pound of water at the temperature of saturated steam at exhaust pressure, in B.t.u. per pound from steam tables.

Engine Efficiency.—The engine efficiency is the ratio obtained by dividing the heat equivalent of the actual work done by the heat available for an ideal engine. The **accepted standard** for the ideal steam engine is the **Rankine cycle** (page 368). The engine efficiency is obtained by the following equation:

$$\text{Engine efficiency (referred to indicated horsepower)} = \frac{2,545}{w(h_{g1} - h_{g3})},$$

¹ See p. 74.

where w = pounds of steam as supplied per indicated horsepower-hour.

h_{g1} = total heat per pound of steam at the initial conditions prevailing before the throttle valve.

h_{g3} = total heat per pound of steam after adiabatic expansion from initial conditions to the final pressure.

h_{g1} and h_{g3} can be found from any steam tables or total heat-entropy or Mollier diagram.¹ $h_{g1} - h_{g3}$ is the heat available for work per pound of steam.

Nominal Cut-off.—The nominal cut-off can be found in the following manner from an indicator card: Through the point of maximum pressure

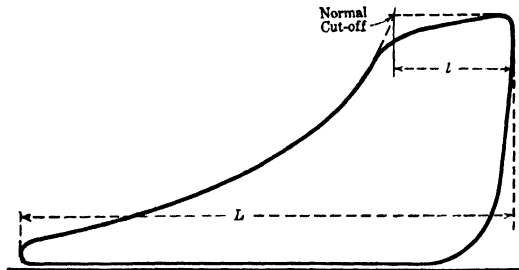


FIG. 267. —Indicator diagram illustrating the point of cut-off.

during admission a line is drawn parallel to the atmosphere line. Then through a point near the top of the expansion line that curve is extended upward to meet the maximum pressure line. The intersection of these two lines is the point of nominal cut-off, and the **proportion of cut-off** is found by dividing the length measured on the diagram up to this point by the total length, as shown in Fig. 267.

The calculation of the steam accounted for by the indicator card has been omitted from the test codes for the following reasons: (a) It is not essential to a performance test of an engine, and (b) as this method does not even approximate the actual steam consumption of an engine, it is liable to be misleading unless thoroughly understood. Methods of calculating this are given on page 368.

TABLE XVIII.—DATA AND RESULTS OF RECIPROCATING STEAM-ENGINE TEST

Short Tests

1. Date.....	
2. Dimensions of cylinders.....	in.
(a) Diameter of piston and tail rods.....	in.
3. Duration.....	hr.
4. Pressure in steam pipe near throttle by gage.....	lb. per sq. in.
5. Pressure in receivers by gage.....	lb. per sq. in.
6. Vacuum in condenser.....	in. mercury at 32°F.
7. Percentage of moisture in steam near throttle or number of degrees of superheat.....	per cent or°F.

¹ See Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 5th ed., pp. 102 and 188, 1933.

- 8. Total steam consumed per hour.....lb.
- 9. Total dry and saturated steam or superheated steam consumed per hour....lb.
- 10. Steam consumed conforming to conditions agreed upon per hour.....lb.
- 11. Average mean effective pressure in each cylinder.....lb. per sq. in.
- 12. Revolutions per minute.....r.p.m.
- 13. Indicated horsepower developed.....i.hp.
- 14. Brake horsepower developed.....b.hp.
- 15. Actual steam rate per indicated horsepower-hour (or brake horsepower-hour) lb.
- 16. Dry steam or superheated steam rate per indicated horsepower-hour (or
brake horsepower-hour).....lb.
- 17. *Corrected* steam rate per indicated horsepower-hour (or brake horsepower-
hour) conforming to conditions agreed upon.....lb.
- 18. Engine efficiency (referred to Rankine cycle) based on indicated horse-
power or brake horsepower.....per cent

TABLE XIX.—DATA AND RESULTS OF RECIPROCATING-STEAM-ENGINE TEST

A.S.M.E. Complete Code (1926)

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

Description, Dimensions, Etc.

- 7. Type of engine (simple or multiple expansion).....
- 8. Class of service (mill, marine, electric, etc.).....
- 9. Rated power of engine.....
- 9a. Auxiliaries (steam or electric drive).....
- 10. Cylinders
 - (a) Diameter.....in.....in.....in.....
 - (b) Cylinder ratio (based on *net* piston displacement) 1 to.....

First Second Third
- 11. Pistons
 - (a) Diameter.....in.....in.....in
 - (b) Diameter of tail rods.....in.....in.....in
 - (c) Stroke.....in.....in.....in
- 12. Clearance volume in per cent of piston displacement:

First Second Third

 - (a) Clearance head end.....per cent..per cent..per cent
 - (b) Clearance crank end.....per cent..per cent..per cent
 - (c) Clearance average.....per cent..per cent..per cent
- 13a. Head end horsepower constant (stroke × net piston area ÷ 33,000).....
- 13b. Crank end horsepower constant (stroke × net piston area ÷ 33,000).....
- 14. Area of interior steam surface (p. 345).....sq. ft.
- 15. Area of jacketed surfaces.....sq. ft.
- 16. Auxiliaries:
 - (a) Valves, type.....
 - (b) Governor, type.....
 - (c) Condensing equipment:

Type.....

Make.....

Rated capacity.....

(d) Pumps (direct or independently driven):

- Oil pump, type.....
- Jacket pump, type.....
- Reheater pump, type.....Size.....
- 17. Driven unit.....
 - (a) Type.....
 - (b) Size.....
 - (c) Rating.....
 - (d) Capacity of generator or other apparatus consuming power of engine.....hp. or kw.
- 18. Run number.....
- 19. Duration of test.....hr.
- 20. Average load.....hp. or kw.
- 21. Initial steam pressure.....lb. per sq. in. abs.
- 22. Initial steam temperature.....°F.
- 23. Temperature of saturated steam corresponding to pressure in exhaust pipe near engine.....°F.
- 24. Exhaust pressure.....lb. per sq. in. or in. mercury at 32°F. abs.
- 25. Actual steam rate.....lb.
- 26. Steam rate per b.hp.-hr. corrected to conditions agreed upon (Item 52 ÷ Item 68).....lb.
- 27. Guaranteed steam rate.....lb. per hp.-hr. or lb. per kw.-hr.
- 28. Heat rate, actual.....B.t.u. per hp.-hr. or B.t.u. per kw.-hr.
- 29. Thermal efficiency, referred to b.hp. per cent
- 30. Engine efficiency (referred to Rankine cycle) based on b.hp. (2,545 ÷ Item 82) × 100.....per cent

Average Pressures

- 31. Barometric pressure.....in. of mercury at 32°F.....lb. per sq. in. abs.
- 32. Pressure in steam pipe near throttle by gage.....lb. per sq. in.
 - (a) Corresponding absolute pressure.....lb. per sq. in.
 - (b) Minimum pressure above atmosphere, steam-pipe diagram near throttle, by gage.....lb. per sq. in.
- 33. Pressure in first receiver, by gage.....lb. per sq. in.
 - Corresponding absolute pressure.....lb. per sq. in.
- 34. Pressure in second receiver, by gage.....lb. per sq. in.
 - Corresponding absolute pressure.....lb. per sq. in.
- 35. Vacuum in exhaust pipe near engine by mercury manometer.in. of mercury at 32°F.
 - Corresponding absolute pressure.....lb. per sq. in.
- 36. Pressure in jackets and reheaters, by gage.....lb. per sq. in.

Temperatures

- 37. Engine-room temperature.....°F.
- 38. Temperature of steam near throttle.....°F.
- 39. Temperature of saturated steam at throttle pressure.....°F.
- 40. Temperature of steam if superheated:
 - (a) Entering first receiver.....°F.
 - (b) Leaving first receiver.....°F.
- 41. Temperature of steam if superheated:
 - (a) Entering second receiver.....°F.
 - (b) Leaving second receiver.....°F.
- 42. Temperature of saturated steam corresponding to pressure in exhaust pipe near engine.....°F.

43. Temperature of steam in exhaust pipe, as observed.....°F.

Quality of Steam at Throttle

44. Number of degrees of superheat or,.....°F.

45. Percentage of moisture in steam.....per cent

Total Steam Quantities

46. Total steam consumed during run by engine, as measured.....lb.

47. Total dry and saturated steam or superheated steam consumed.....lb.

48. Correction factor conforming to conditions agreed upon (p. 360).....lb.

49. Total steam consumed, corrected to conform to conditions agreed upon
(Item 46 $\times$ Item 48).....lb.

Hourly Quantities

50. Total steam consumed per hr. as measured (Item 46 $\div$ Item 19).....lb.

51. Dry and saturated steam or superheated steam consumed per hour (Item
47 $\div$ Item 19).....lb.

52. Total steam consumed per hr. corrected to conditions agreed upon (Item 49 $\div$
Item 19).....lb.

Heat Consumption

53. Heat content per pound of steam at throttle.....B.t.u.

54. Heat content of liquid at temperature of steam at exhaust pressure.....B.t.u.

55. Heat supplied per pound of steam.....B.t.u.

56. Heat consumed per hour (see Calculation of Heat Rate (p. 360).....B.t.u.

57. Net work per pound of steam, from adiabatic expansion in ideal engine
operating between the same initial conditions and final pressure according
to Rankine cycle (see p. 360).....B.t.u.

Indicator Diagrams

	First Cyl.	Second Cyl.	Third Cyl.
58. Nominal cut-off, per cent.....			
59. Mean effective pressure (average), lb. per sq. in.....			
60. Maximum pressure above atmosphere, lb. per sq. in.....			
61. Absolute back pressure at lowest point, lb. per sq. in.....			

Speed

62. Revolutions per minute.....r.p.m.

(a) Variation of speed between no load and full load.....per cent

(b) Momentary fluctuation of speed on suddenly changing from full load
to half load.....per cent

63. Piston speed.....ft. per min.

Power—indicated horsepower

64. First cylinder, crank end.....i.hp.
head end.....i.hp.

65. Second cylinder, crank end.....i.hp.
head end.....i.hp.

66. Third cylinder, crank end.....i.hp.
head end.....i.hp.

67. Indicated horsepower developed by entire engine.....i.hp.

68. Brake horsepower developed by entire engine.....b.hp.

69. Friction of engine (Item 67—Item 68).....hp.

70. Percentage of friction (Item 69 $\div$ Item 67) $\times$ 100.....per cent
 71. Indicated horsepower with no load at normal speed.....i.hp.
 72. Mechanical efficiency (Item 68 $\div$ Item 67) $\times$ 100.....per cent

Economy Results

73. Steam rate per indicated horsepower-hour as measured (Item 50 $\div$ Item 67).....lb.
 74. Dry and saturated or superheated steam consumed per indicated horsepower-hour (Item 51 $\div$ Item 67).....lb.
 75. Steam rate per indicated horsepower-hour corrected to conform to conditions agreed upon (Item 52 $\div$ Item 67).....lb.
 76. Steam rate per brake horsepower-hour as measured (Item 50 $\div$ Item 68).....lb.
 77. Dry and saturated steam or superheated steam consumed per brake horsepower-hour (Item 51 $\div$ Item 68).....lb.
 78. Steam rate per brake horsepower-hour corrected to conform to conditions agreed upon (Item 52 $\div$ Item 68).....lb.
 79. Heat rate per indicated horsepower-hour (Item 56 $\div$ Item 67).....B.t.u.
 80. Heat available according to Rankine cycle per indicated horsepower-hour (Item 57 $\times$ Item 73).....B.t.u.
 81. Heat rate per brake horsepower-hour (Item 56 $\div$ Item 68).....B.t.u.
 82. Heat available according to Rankine cycle per brake horsepower-hour (Item 57 $\times$ Item 77).....B.t.u.

Efficiency Results

83. Thermal efficiency referred to indicated horsepower (2,545 $\div$ Item 79) $\times$ 100.....per cent
 84. Thermal efficiency referred to brake horsepower (2,545 $\div$ Item 81) $\times$ 100 per cent
 85. Engine efficiency (referred to Rankine cycle) based on indicated horsepower (2,545 $\div$ Item 80) $\times$ 100. See p. 360.....per cent
 86. Engine efficiency (referred to Rankine cycle) based on brake horsepower (2,545 $\div$ Item 82) $\times$ 100.....per cent

Specimen Diagrams

87. Sample diagram from each cylinder.....
 88. Sample steam-pipe diagram.....

NOTE 1.—When the *engine drives an electric generator*, the following additional data and results applicable especially to alternating-current generators should be entered in the table.

Electrical Data

89. Average volts, each phase.....volts
 90. Average amperes, each phase.....amp.
 91. Power factor.....per cent
 92. Total output in kilowatts.....kw.
 93. Net output in kilowatts (see Calculation of Electrical Horsepower, p. 378).....kw.
 (a) Field volts.....volts
 (b) Field amperes.....amp.

Power and Economy

94. Electrical horsepower developed (Item 93 $\div$ 0.7457).....hp.
 95. Steam rate per net kilowatt-hour (Item 50 $\div$ Item 93).....lb.
 96. Dry and saturated steam or superheated steam consumed per net kilowatt-hour (Item 51 $\div$ Item 93).....lb.

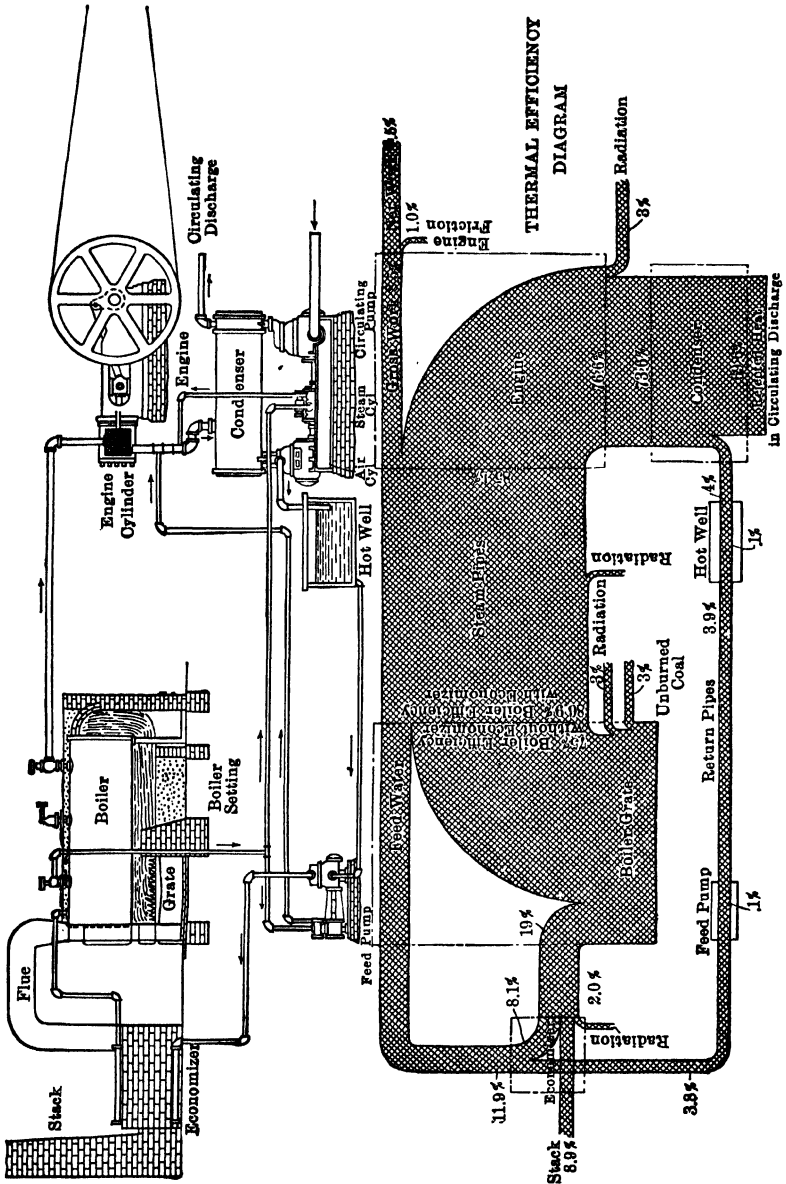


FIG. 268.—Heat distribution and losses in a steam boiler and engine plant.

97. Corrected steam rate per net kilowatt-hour conforming to conditions agreed upon (Item 52 ÷ Item 93).....lb.
 98. Heat rate per net kilowatt-hour (Item 56 ÷ Item 93).....B.t.u.

NOTE 2.—When testing a *marine engine* having a shaft dynamometer (p. 178), the form should include the data obtained from this instrument, in which case the brake horsepower becomes the *shaft horsepower*.

Hirn's Analysis.—For certain scientific investigations it is useful to make a heat analysis of the indicator diagram, to show the interchange of heat from steam to cylinder walls, etc., which is going on within the cylinder. This is unnecessary for commercial tests. Years ago deductions from such an analysis were considered to be of considerable importance to designers, but lately such data are considered of very doubtful importance.¹

Figure 268 shows diagrammatically the heat distribution and the various losses as percentages in a typical small steam plant.

Ratio of Economy of an Engine to That of an Ideal Engine.—The cycle of the ideal engine recommended for obtaining this ratio is that which was adopted many years ago by the Committee appointed by the *Civil*

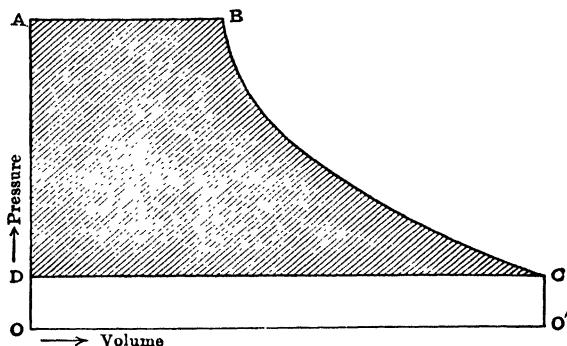


FIG. 269.—Indicator diagram for the Rankine cycle.

Engineers of London, to consider and report a standard thermal efficiency for steam engines. This engine is one which follows the **Rankine cycle** where steam at a constant pressure is admitted into a cylinder having no clearance, and after the point of cut-off is expanded adiabatically to the back pressure. In obtaining the economy of this engine the feed water is assumed to be returned to the boiler at the exhaust temperature (Fig. 269).

The ratio of the economy of an engine to that of the ideal engine is obtained by dividing the heat consumption per indicated horsepower per hour for the ideal engine (called the "theoretical water rate," see page 377) by that of the actual engine.

¹ For more information about Hirn's analysis and other heat-analysis methods, see Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 3d ed., p. 144, 1925.

Engine Performance Compared with the Rankine Cycle.—In order to know how much the efficiency of an engine can be improved, it is most desirable to compare the actual thermal efficiency with the highest possible efficiency. For steam engines the standard cycle for comparison is now generally taken as the Rankine cycle,¹ in which the operation of the engine is assumed to be perfect, that is, without clearance in the cylinder, initial condensation, leakage, or radiation. The indicator diagram for the Rankine cycle is represented by Fig. 269. The steam is assumed to be supplied to the engine cylinder at constant pressure until the point of cut-off, after which it is expanded **adiabatically** down to the back pressure at which the engine is operated on the return stroke when the exhaust steam is swept out of the cylinder and returned as feed water to the boiler at the temperature of the exhaust. The same Rankine cycle represented in Fig. 269 when shown by a so-called **entropy-temperature diagram** can be made simpler for both analysis and calculations. This other kind of diagram, the details of which are somewhat more difficult to understand, is universally used by steam-turbine engineers and has for the problem in hand particular advantages. In this diagram, any surface represents accurately to given scales a quantity of heat. Absolute temperatures T are the ordinates, and entropies² ϕ are the abscissas.

Specific Heat of Superheated Steam.—In modern practice superheated steam often enters our calculations. The specific heat of steam varies with the temperature and pressure as shown in Fig. 270 giving values of the **mean** specific heat at constant pressure (C_p).³

Approximate Steam Consumption Calculated from an Indicator Diagram.—It is often very convenient to be able to calculate the approximate steam consumption of a steam engine from the data obtainable from an indicator card, the size of the piston, the stroke, and the speed. In using a double-acting engine, the following symbols⁴ may be used:

p = mean effective pressure, in pounds per square inch from indicator diagram.

¹ Years ago it was not unusual to make this comparison with the efficiency of the Carnot cycle as a basis. This efficiency of a heat engine, it will be remembered, is expressed by the ratio of $T_1 - T_2$ to T_1 , where T_1 is the absolute initial temperature and T_2 is the absolute final temperature.

² Entropy, which Perry calls the "ghostly quantity," has no real physical significance, so that complete definition is not possible. If dQ is a small amount of heat added to a body and T is the absolute temperature at which the heat is added, then the change in entropy of that body is dQ/T , or $d\phi = dQ/T$. Entropy of saturated steam above the entropy of water at the freezing point is easily calculated. For saturated steam at any pressure, then $\phi = xh_{fg}/T + s_f$, where x is the quality of the steam, h_{fg} is the heat of evaporation, T is the absolute temperature, and s_f is the entropy of the liquid (water).

³ These data vary in some values slightly from those used in calculating Keenan's steam tables (A.S.M.E. standard), given in the Appendix (Tables I, II, and III).

⁴ Compare with *Power*, September, 1893.

- l = length of the stroke of the engine, in feet.
 a = area of the piston, in square inches.
 b = percentage of clearance to the length of the stroke.
 c = percentage of stroke at any point in the expansion line.¹
 n = number of revolutions per minute; and $120n$ = number of strokes per hour.
 w = weight of a cubic foot of steam having a pressure as shown by the indicator diagram corresponding to that at the point in the expansion line selected for c , in pounds.
 w' = weight of a cubic foot of steam corresponding to the pressure at the end of compression, in pounds.

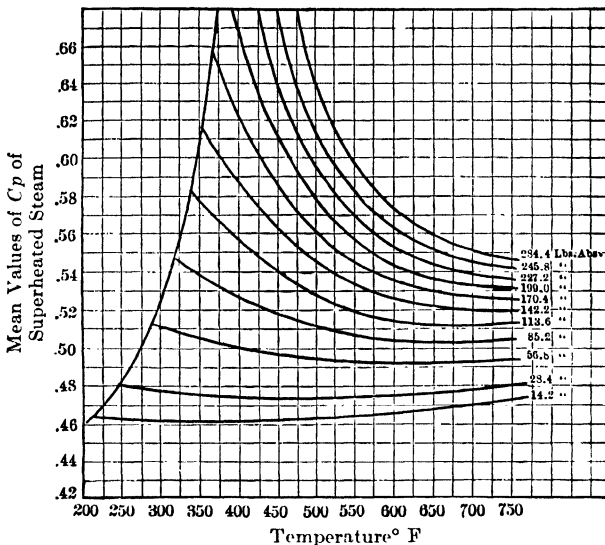


FIG. 270.—Mean values of specific heat (C_p) of superheated steam.

Then the number of cubic feet per stroke = $\frac{la(b+c)}{144(100)}$ in the clearance and piston displacement volumes (at c).

$$\text{Weight of steam per stroke, in pounds} = \frac{law(b+c)}{144(100)}. \quad (80)$$

$$\text{Volume of the clearance, in cubic feet} = \frac{la(b)}{144(100)}.$$

¹ In other words, this is the percentage of the entire stroke which has been swept through by the piston corresponding to the point in the expansion curve selected for measurements. It is preferable, however, to take this point not very far from the point of cut-off, since the assumption must be made that the product of pressure times volume in the expansion curve is constant, which is, of course, not accurate, and the error becomes greater as the expansion increases.

Weight of steam in clearance, in pounds remaining in the cylinder

$$= \frac{law'(b)}{144(100)}.$$

Approximate net weight of steam used per stroke

$$= \frac{law(b+c)}{144(100)} - \frac{law'(b)}{144(100)} = \frac{la}{14,400}[(b+c)w - bw']. \quad (81)$$

Approximate weight of steam from diagram per hour

$$= \frac{120 nla}{14,400}[(b+c)w - bw']. \quad (82)$$

Indicated horsepower for a double-acting engine

$$= \frac{2 \text{ plan}}{33,000}. \quad (83)$$

Steam consumption per indicated horsepower¹ is Eq. (82) divided by Eq. (83) or

$$= \frac{137.5}{p}[(b+c)w - bw']. \quad (84)$$

The difference between the theoretical steam consumption calculated by the formula and the actual consumption as determined by tests represents steam "not accounted for by the indicator," due to cylinder condensation, leakage through ports, radiation, etc. If the steam supplied to the engine is very wet, corrections for this moisture should be made in the value of w .

In the 1926 A.S.M.E. Power Test Codes for Reciprocating Steam Engines (see page 361) it is stated that the method of calculating steam accounted for by the indicator card "does not even approximate the actual steam consumption of the engine" and is therefore likely to be misleading. For this reason no larger amount of space is given here.

Willans Law.—One of the most serviceable **checks** that can be applied to engine tests is plotting the **Willans line** of total steam consumption per hour. Curve sheets illustrating this as plotted from actual tests by Barraclough and Marks² are shown in Fig. 271. It will be observed that the points representing the weight of steam used per hour when plotted for the horsepower corresponding are on a **straight line**. In other words, Willans law is usually stated thus: "With a fixed cut-off and a **throttling**

¹ A method of determining steam consumption by means of logarithmic curves of indicator diagrams by J. P. Clayton is described in *Bull. 65 of Engineering Experiment Station of University of Illinois* and in abridged form in Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 3d ed., p. 147; also in *Jour. A.S.M.E.*, April, 1912.

² *Proc. Inst. Civil Eng.*, Vol. 120, p. 323.

governor the total steam used by the engine per hour at different loads can be represented by a straight line upon a mean effective pressure base or upon a horsepower base." It will be shown also in the following paragraphs how, theoretically, this relation holds for an engine operating at a fixed cut-off and with a throttling governor and that the total steam used per hour is proportional to the mean effective pressure and also to the horsepower developed.

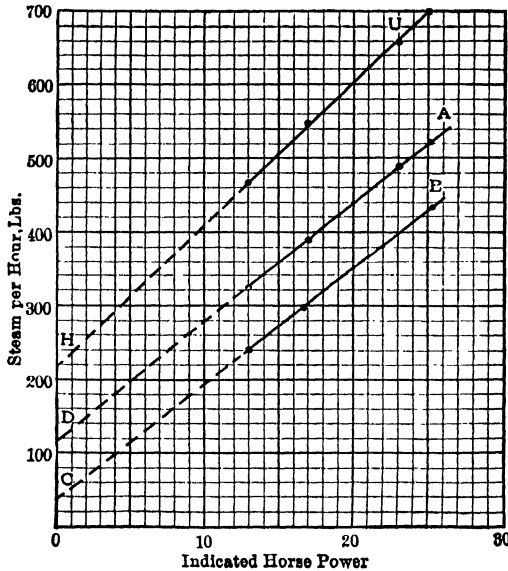


FIG. 271.—“Willans lines” for an engine with a throttling governor.

If we assume that the expansion curve is hyperbolic, which is usually near the truth, then the mean “forward” pressure given by an indicator diagram is¹

$$p_1 \left(\frac{1 + \log_e r}{r} \right). \quad (85)$$

where p_1 is the initial pressure of the steam and r is the ratio of expansion. With a **throttling** governor r is, of course, constant. The terms in the parenthesis can then be represented by a constant c and the mean forward pressure p_f then can be written as $p_1 c$. Volume of steam used per hour for a **double-acting** engine can be expressed in cubic feet as $120(nV)$, where n is the number of revolutions per minute and V is the volume of steam admitted to the cylinder per stroke. Now if we use the symbol v_1 for the specific volume, that is, the volume in cubic feet of a pound of steam at the pressure p_1 , and assume for a small range of pressures

¹ Compare with Perry, “Steam Engine,” p. 286.

that $p_1 v_1 = \text{a constant } k$, then we can write, if W is the weight of steam used per hour in pounds,

$$W = \frac{120(nV)}{v_1} = \frac{120(nVp_1)}{k}. \quad (86)$$

Now with a throttling governor and constant cut-off all these quantities are constant except p_1 , and writing a constant z for the term $\frac{120(nV)}{k}$ we have $W = zp_1$, but it was shown above that the mean **forward** pressure $p_f = cp_1$, so that $W = z\frac{p_f}{c}$.

Therefore the curve representing this equation is a **straight line** and passes through the origin of coordinates. If, however, we use the mean **effective** pressure instead of the mean forward pressure, then

$$\text{m.e.p.} = p_f - p_b,$$

where p_b is the mean "back" or exhaust pressure. In these last terms then

$$W = \frac{z}{c}(\text{m.e.p.} + p_b). \quad (87)$$

This last equation may be stated as $W = \text{a constant} \times \text{m.e.p.} + \text{another constant}$ which, when plotted to a scale of mean effective pressure for abscissas and weight of steam used per hour for ordinates, is also a **straight line**, intersecting the axis of ordinates above the origin at a distance corresponding to the steam consumption per hour at no load.

Since the **indicated horsepower** (i.hp) is proportional to mean effective pressure, a straight line will result when the steam consumption and indicated horsepower are plotted; and the same holds true also when steam consumption is plotted with **brake horsepower** (b.hp.) instead of indicated horsepower.

Curves Showing Results of Tests Graphically.—One of the best checks of an engine test is to plot the principal observations graphically as the test proceeds. This is particularly important as regards the **total weight of steam used per hour**. For a series of tests each made with a different load the points plotted with horsepower (either indicated or brake) as abscissas and total steam per hour as ordinates should lie along a **straight line**, known as the **Willans line**. This statement applies accurately only for engines operating with a throttling governor at, of course, constant speed but is generally applicable to steam turbine tests (page 375), irrespective of the type of governor.

If steam-rate tests have been run at several loads, the principal results will be shown in curve form. Such curves should all be plotted with a common scale of load as abscissas, with the following ordinates: (1)

total steam, to as large a scale as possible; (2) steam rate, or heat consumption; (3) thermal efficiency; (4) engine efficiency.

As a check on operating conditions, the report should present for each test run graphic logs, with time as abscissas, and with the following ordinates: (1) initial steam pressure; (2) initial steam temperature or moisture; (3) exhaust pressure; (4) load; (5) rate of flow of condensate.

NOTE.—Power Test Codes have been prepared by the A.S.M.E. on the following subjects, which for lack of space could not be included in this book. Subjects and references are given below:

Code for Evaporative Apparatus (see *Mech. Eng.*, March, 1922).

Code for Locomotives (see *Mech. Eng.*, October, 1923).

Code for Hydraulic Power Plants (see *Mech. Eng.*, April, 1922).

CHAPTER XIII

TESTING STEAM TURBINES AND TURBOGENERATORS

Testing Steam Turbines.¹—In every power plant the means should be available for making tests of the steam equipment to determine the steam consumption. Usually tests are made to determine how nearly the performance of a turbine approaches the conditions for which it was designed. The results obtained from tests of a turbine are to show usually the steam consumption required to develop a unit of power in a unit of time, as, for example, a horsepower-hour or a kilowatt-hour.

The most valuable test of a steam turbine or of a reciprocating steam engine is made when varying only the load, that is, with pressures, superheat, and speed constant. When the steam consumption is then plotted against fractions of full load, a **water-rate curve** is obtained.² For such a curve a series of tests is needed, each for some fraction of full load; and in each separate test the power as well as all the other conditions must be held constant.

Another important test of the performance of a steam turbine is made by varying both the speed and the power and keeping the other conditions constant. The observations of speed and power from such a test give a power parabola as illustrated in Fig. 272. This curve shows at what speed the turbine gives the greatest output.

Methods for Testing.—The important observations to be made in steam turbine tests are:

1. Pressure of the steam supplied to the turbine.
2. Speed of rotation of the turbine shaft.
3. Measurement of power with a brake or with electrical instruments.
4. Weight, or measurement by volume, of the condensed steam.

¹ Tests of the turbines alone in a modern station may be only a rough indication of the overall economy of the plant. Recently steam turbines were installed in a large power plant where they replaced steam engines of an excellent make. Tests of the turbines and of the engines made without considering the losses in the rest of the plant showed very little gain in efficiency by this change, although it was found that the fuel consumption was reduced 20 per cent.

Parts of this chapter and the chapter following are taken from the author's work on "Steam Turbines," 6th ed., published by John Wiley & Sons, Inc., New York.

² The most satisfactory tests of turbines are made with steam slightly superheated rather than wet. When steam is very wet (more than about 4 per cent moisture for ordinary pressures), the determination of the quality is difficult. There is also a danger that steam showing only a few degrees of superheat by the reading of the thermometer is actually wet. The high temperature is due in such cases to heating from eddies around the thermometer case or in steam pockets near it.

Tests may also be made with varying initial steam pressure but with other conditions, including exhaust pressure and load, constant.

Calculations of the steam consumption and efficiency of turbines made by allowing for the different losses as calculated separately and then added together, as is often done to determine the losses in electrical apparatus, are of very little value except when made by experienced designers.¹

It has sometimes happened that split condenser tubes have caused a leakage of steam which was extremely difficult to measure. Cases are reported where the split opened up only when the condenser was heated with a large volume of steam. On this account it is preferable not to use a leaky condenser for accurate tests; in other words, the condenser should be thoroughly repaired before tests are made. The effect of split tubes causing an irregular amount of leakage is usually shown in tests by inconsistent results in the weight of condensed steam. In that case the leakage will be greatest with largest flow of steam through the condenser and it will be observed, for an engine operating with a throttling governor or with a steam turbine, that when the "Willans line" (page 372) is drawn to check the tests, it will be curved instead of straight. It should be noted, however, that a curved "Willans line" does not necessarily indicate condenser leakage, as the irregularity may be due to faulty design of the engine² or turbine.

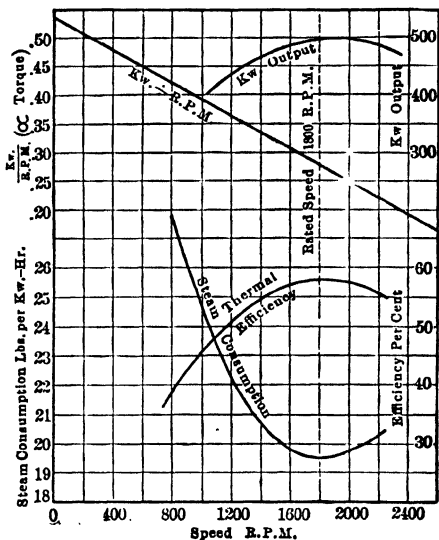


FIG. 272.—Results of tests of a turbine at various speeds.

Steam Consumption Determined by a Heat-balance Method.—Still another method is sometimes used for determining the steam consumption of engines and turbines operated with jet condensers or condensers of a similar type where the cooling water and condensed steam are mixed and discharged together from the condenser. This method is based on the measurement of the amount of heat absorbed by the

¹ This method of calculation of steam consumption is explained in detail in "Steam Turbines," by the author, 6th ed., pp. 86–93. Steam consumption of a turbine can be predicted by calculations much more accurately than for a steam engine.

² The "Willans" line for a reciprocating engine operating with an automatic cut-off governor is usually a curve slightly concave upward.

cooling water from the condensed steam. The weight w_c and temperature of the cooling water leaving the condenser t_2 , the quality of the exhaust steam x , and the temperature of the mixture of condensed steam and cooling water t'' are determined as accurately as possible and from these data the weight of condensed steam w_s is, of course, readily calculated by a simple algebraic equation as follows:

$$w_c(t_2 - 32) + w_s(h_f + xh_{fg}) = (w_c + w_s)(t'' - 32), \quad (88)$$

where h_f and h_{fg} are, respectively, the heat of the liquid and the heat of vaporization corresponding to the temperature of the exhaust steam. In this equation "heat contents" are measured for each term from 32°F. The method is, however, unreliable and at best can be depended on for only very approximate results. The reason for this inaccuracy is the difficulty of measuring, especially in large plants, the quantity of cooling water and the true average temperature of a large volume of water flowing in a pipe or channel. It is found usually that the temperature of the water discharged from the condenser will vary from one side of the pipe to the other, and small errors in the determination of this temperature, because the rise in temperature is small, will make large discrepancies in the calculated weight of condensed steam.

Steam Consumption of Auxiliaries by Calculation of Heat Balance.—

The heat-balance method of the preceding paragraph can be adapted also to the calculation of the steam used by noncondensing auxiliaries discharging their exhaust into a feed-water heater, provided, of course, all the steam entering the heater is condensed.

W = weight of condensed steam from turbine, in pounds.

w_o = weight of make-up water, in pounds.

w_a = weight of steam used by auxiliaries, in pounds.

t_1 = temperature of water entering heater, in degrees Fahrenheit.

t_2 = temperature of water leaving heater, in degrees Fahrenheit.

t_o = temperature of make-up water, in degrees Fahrenheit.

t_a = temperature of steam corresponding to back pressure, in degrees Fahrenheit.

h_{fga} = heat of vaporization, corresponding to back pressure, in B.t.u. per pound.

$$W(t_2 - t_1) + w_o(t_2 - t_o) = w_a(h_{fga} + t_a - t_2). \quad (89)$$

Stage pressures should be observed and recorded when tests are made of steam turbines having a series of pressure stages. These data are often extremely useful, both for checking the weight of condensed steam if the turbine is of the nozzle type¹ and also for showing any abnormal conditions in the several stages.

¹ Usually the nozzles discharging the steam into the second stage of the turbine are always open, so that the total area is always constant. If, therefore, the areas of the smallest sections of these nozzles are measured and the pressure is observed in the first stage, the weight can be calculated with a considerable degree of accuracy by using the formula for the flow of steam on p. 195.

Results calculated on a basis of kilowatts output should be *net*; that is, the power required for excitation should be subtracted from the generator output. If, however, the generator is self-exciting, the net output can be measured directly at the terminals of the machine.

Different machines will have different correction factors for varying conditions of pressure, superheat, vacuum, etc., so that water rates corrected for large variations are always likely to be more or less inaccurate. This is particularly true in respect to vacuum corrections. Some turbines will give a very good efficiency with a low vacuum, but at a high vacuum because of an insufficiently large steam space the efficiency will be low.

CALCULATIONS OF EFFICIENCY (SHAFT AND BUCKET) OF A STEAM-TURBINE GENERATOR
COMPARED WITH THE RANKINE CYCLE

1. "Electrical" kilowatts.
 2. R.p.m.
 3. Steam per hour (corrected for moisture).
 4. Water rate per "electrical" kilowatt, in pounds per hour. Item 3 ÷ Item 1.
 5. I^2R , loss in generator, in kilowatts.
 6. Rotation loss of generator alone, in kilowatts.
 7. Rotation loss of wheel and generator, in kilowatts.
 8. "Shaft" kilowatts. Item 1 + Item 5 + Item 6.
 9. "Bucket" kilowatts. Item 1 + Item 5 + Item 7.
 10. Water rate per "shaft" kilowatt, in pounds per hour.
 11. Water rate per "bucket" kilowatt, in pounds per hour.
 12. Steam-chest pressure, in pounds per square inch, absolute.
 13. Exhaust pressure, in pounds per square inch absolute.
 14. Available energy, in B.t.u. per pound of steam.
 15. *Theoretical water rate*, in pounds per kilowatt-hour = $\frac{3,413}{\text{available energy (Item 14)}}$.
- (See page 74.)
16. "Shaft" efficiency = Item 15 ÷ Item 10.
 17. "Bucket" efficiency = Item 15 ÷ Item 11.

NOTES.—For calculating rotation loss of a new design, stage pressures are, of course, used.

Steam per hour is usually calculated from the area of the nozzles in the first stage if the governor is *not operating*. For a speed-torque test the *flow of steam* is constant and kilowatts for determining Items 8 and 9 are read from this curve $\left(\text{kw.} = \frac{\text{kw.}}{\text{r.p.m.}} \times \text{r.p.m.} \right)$.

The speed-output curve (Fig. 272, page 375) is very useful to engineers in determining if a turbine is running at its best speed. If the corresponding curves of steam consumption per kilowatt output (usually called water rate per kilowatt) and **efficiency** curves are calculated according to the foregoing form, a great deal of information is obtained about the operation and economy of a turbine. The **torque line** in Fig. 272 is always drawn straight, just as a "Willans line." A curve of **total** steam consumption is usually a straight line for the normal operating limits of a turbine but usually becomes curved when a by-pass valve opens on over-

load, or when the turbine is over its capacity so that the pressures are not normal in the stages.

The torque line shows why a **turbine engine is not adaptable to automobiles**. The starting torque of a small commercial turbine is not large, so that starting would be difficult with a small wheel, and reversing and speed reduction would be as difficult as with a gasoline engine. The reciprocating steam engine as well as the gasoline engine has, therefore, advantages over the steam turbine for this service. Steam turbines can, however, be used successfully for the motive power of **airplanes**.

Electrical Output of Turbine Generators. Measurement of Direct Current.—Careful engineers will not ordinarily use the instruments on the switchboard of a power station for measuring the electrical output of a generator, because, unless exceptional precautions have been taken to avoid “stray” magnetic fields and the instruments have been calibrated in place under operating conditions with a **sufficient interval of time between observations** of current (amperes) at different loads so that the **shunts of the ammeters will reach a constant temperature** for the particular value of current flowing, there may be considerable error in the observations. Switchboard voltmeters are usually satisfactory if they are carefully calibrated; but the shunts of the type of ammeters ordinarily used have a drop of approximately only 60 millivolts, so that the indicating part of the ammeter must be almost entirely a circuit of copper wire. It is for this reason that such instruments are likely to be affected considerably by varying room temperatures, and with some shunt arrangements they are susceptible to errors also from variations in the value of the circuit itself. For accurate measurements, it is therefore best to use only the **portable** types of indicating ammeters having shunts with a drop of 200 millivolts.¹ In these latter instruments the indicating part is made up largely of resistance wires having practically no temperature coefficient. Portable voltmeters are also to be preferred to those on the switchboards.

Unless standard shunts having a drop of 200 millivolts as provided for good portable ammeters are used, the influence of “stray” magnetic fields must be guarded against. When, on the other hand, switchboard instruments are used, such influences must be investigated and arrangements must be devised so that “stray” fields will not affect the measurements. The influence of very weak magnetic fields can be eliminated from the final results by turning the instruments between successive readings. Observations of current (amperes) made with the switchboard type of instruments are also often in error owing to thermoelectric effects producing a small electromotive force sufficient, however, to alter

¹ This value for the drop in shunts is an arbitrary value selected by a number of makers of electrical instruments because it gives the best compensation of all the temperature errors (see *Gen. Elec. Rev.*, February, 1911).

the readings of the millivoltmeter. The error due to this cause can be observed by reading the millivoltmeter at the close of the test immediately after the current has been shut off in the main circuit. It should be, of course, the object of the observer to take this reading before the shunts and leads have cooled appreciably. If there is an error due to this cause, there will be a small positive or negative deflection of the needle from the correct zero, which should be applied as a correction to all the observations of current.

Measurement of Alternating Current.—The same general precautions outlined above for direct-current instruments must be observed in the use of those for alternating current. Although **steady** magnetic fields are not often a cause of much trouble, it happens often, particularly in the case of large generators, that large magnetic fields are influencing the measuring instruments, which have the same frequency as that of the current measured. To eliminate the effect of such "stray" fields, shielded types of instruments should be used. Only with the most expert handling can accurate results be expected when unshielded instruments are used. For measuring large values of alternating current, instrument transformers are generally used. These should be of the precision type and should be sent to a standardizing laboratory, before and after a series of tests, to be calibrated, and a certificate of accuracy should be obtained. The transformers should be calibrated at as nearly as possible the values of the current to be measured in the tests.

Whenever it is possible, tests of generators should be made with a non-inductive load, water rheostats being usually the most satisfactory apparatus for providing such a load. At least, tests should be made under conditions giving a low power factor, so that there can be no error in the readings of the instruments due to phase displacements in the instrument transformers. With a purely noninductive load the readings of the ammeters and the voltmeters can be used to check the wattmeters. Although the **readings of the wattmeters should be taken as the correct value of the output**, the apparent power as indicated by the ammeters and voltmeters should agree with the wattmeter readings within 1 per cent. If a noninductive load cannot be secured, the switchboard ammeters and voltmeters will be satisfactory for readings, to indicate whether or not the load on the circuits is properly balanced. **Watt-hour meters** are not usually satisfactory for the accuracy expected in most tests, and the use of these instruments should be generally avoided. It is only in the case where tests must be made under extremely variable service conditions, where it is difficult to obtain a true average from the readings of indicating instruments, that a watt-hour meter, either for direct or for alternating current, may sometimes give more accurate results than the portable indicating types of instruments. Whenever watt-hour meters are used in tests, they should be checked in place

for a series of constant loads at the frequency, voltage, etc., which are to be used in the test.

Single-phase indicating instruments are to be preferred for measurements of polyphase current to the standard types of so-called polyphase instruments, as the indications of a polyphase instrument are produced by two influences from separate phases of the current in such manner that a correction cannot be applied to obtain true values unless the division of the load is determined by the use of single-phase instruments. Obviously, then, if it is necessary to have single-phase instruments in the separate circuits, it is desirable to have them of the precision type, and polyphase instruments are not needed.

A.S.M.E. TEST CODE (1928) FOR STEAM TURBINES (ABRIDGED)

From the standpoint of steam utilization, steam turbines may be classified as follows:

(a) *Complete-expansion* turbines (no extraction, no reheating, no mixed-pressure features—all steam enters at one pressure, and all leaves at one pressure); (b) *turbines with intermediate reheating* of the steam; (c) *regenerative turbines*, from which steam is *extracted* only for heating feed water, and for which the performance is to be determined on a thermal basis, the turbine being credited with the heat returned to feed water by steam extracted from the turbine; (d) *extraction turbines*, from which steam of unknown quality is extracted for exterior purposes; (e) *mixed-pressure turbines*; (f) *combined mixed-pressure and extraction turbines*.¹

For the present, this code provides instructions for steam-rate or heat-rate tests (page 392) for cases *a*, *b*, and *c* only, with the essential auxiliaries, such as oil pumps and generator fans, whose operation is strictly necessary to the operation of the turbine itself. The determination of the steam rate or the heat rate in cases *d*, *e*, and *f* is a difficult matter, wherefore it is deemed neither feasible nor desirable to reduce such testing to the basis of a general code. The principal difficulty lies in the determination of the quantity or quality of low-pressure steam, either or both of which are required in each of these cases. Such determinations are in the nature of research rather than of testing at the present time. In specific cases conditions may be such that this code may be followed, as, for example, if the low-pressure steam is superheated.

Load on Turbine.—The load on the turbine is the power being developed expressed in brake horsepower or in net kilowatts delivered. In some cases observations of this quantity by indicating instruments serve as the basis for computation of the output, and in such cases the load (or data from which the load may be computed) must be observed at frequent intervals (not ordinarily more frequently than every 5 minutes), and with a high degree of accuracy.

The **momentary load** should not differ more than 5 per cent from the average load. In some cases the rate of steam flow, as indicated by a

¹ See "Steam Turbines," by the author, 6th ed., pp. 311–342.

flow meter or the steam pressure in intermediate stages of a turbine, may be significant in indicating load variations. The load on a turbine running alone may be measured by means of any suitable dynamometer, such as friction brake or torsion meter. The load on a turbogenerator is the net output of the generator (page 378).

Total Steam Used.—The total steam shall be determined as follows:

a. For a turbine exhausting to a surface condenser: by weighing or measuring the condensate.

b. For a turbine exhausting to atmosphere, or to a jet condenser: (1) by weighing or measuring the water fed to a boiler whose only open outlet is the pipe leading to the turbine under test; (2) by removing and measuring flow through the first-stage nozzle block.¹

When water is weighed, scales shall be fitted with tanks arranged so that a continuous flow can be cared for, and the scales shall not show an error greater than 2 in 1,000. When water is measured, the volumetric tanks used shall be accurately calibrated by actual weighing, and a correction shall be determined and applied for water temperature.

In connection with any test in which the total steam is determined by weighing or measuring the condensate from a surface condenser, the condenser must be tested for leakage of cooling water into the condensate (page 356). If this leakage exceeds the limits given below, the condenser is not in condition suitable for a reliable test, and it shall be made satisfactorily tight before the test. These limits depend upon the rated capacity of the turbine and are expressed as a percentage of the rated, or guaranteed, steam flow at rated load.

	Per Cent
Less than 500 kilowatts or 700 horsepower.....	1.00
500 to 1,000 kilowatts or 700 to 1,400 horsepower.....	0.75
Over 1,000 kilowatts or 1,400 horsepower.....	0.50

If **condenser leakage** is determined by silver nitrate (for salt water) (page 349), or by the electrolytic-conductance method (page 355), observations shall be made every half hour during the turbine steam-rate test, in addition to the observations made during the preliminary leakage test.

The water level in the condenser hot well or inlet pipe to the condensate pump shall be checked at the beginning and end of a run, and at intermediate points which mark the division of the run into periods, for each of which the steam rate is to be determined; and correction shall be made for any outstanding change in the quantity of water stored. It is highly desirable that means be employed for holding the

¹ A steam-flow method using standardized nozzles is recommended by the A.S.M.E. Research Committee on Fluid Meters for determining the total steam used by extraction (bleeder) and noncondensing steam turbines. This method is explained on page 197 and more in detail in *Trans. A.S.M.E.*, Vol. 55, No. 11, p. 164, 1933.

condensate level constant throughout the test, as, for example, by throttling the discharge of the condensate pump. One satisfactory way of obtaining the desired result is to note that a gage glass on the inlet pipe to the condensate pump shows a constant level. If the volume of the pipe between the gage glass and the condensate pump is small, it will be sufficient to see that the gage glass is always empty. The resulting error can easily be computed and will usually be negligible. There should be no other points at which water can accumulate. If such

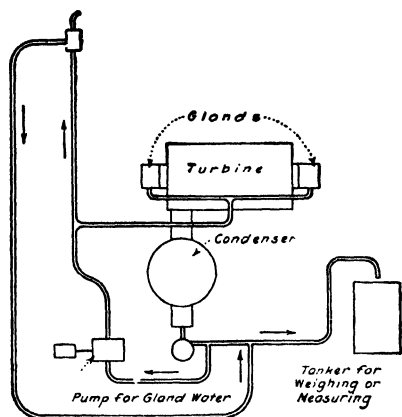


FIG. 273.—Piping when condensate is used for "sealing" water.

places cannot be avoided, proper correction shall be made for each of them just as for the hot well. All lines leading from and to the condenser, other than the air and condensate pump suction and turbine drips, shall be blanked off or fitted with two valves with a bleeder between them.

If condensate is used for any purpose, such as for sealing pump glands, turbine glands, or the atmospheric relief valve, the quantity so used must be determined and included in the total. Condensate lost through leaky valves must be determined and included, if it has not previously been

accounted for. Water other than condensate entering the condenser, or other parts of the system, and weighed or measured condensate reentering the condenser must be determined and deducted from the total. It is plainly desirable that the arrangement of piping, etc., be such that the need of such corrections to the test results be reduced to a minimum. For instance, in turbines equipped with water glands with condensate employed for sealing, they may be piped up as shown in Fig. 273. Water is taken from the line between the condensate pump and weighing tanks and pumped to the desired head for sealing the glands, and the overflow is returned to the same pipe. Of course, any external leakage from such glands must be collected, properly weighed, and charged against the turbine. Similarly, any gland or water seal employed to preclude air leakage to parts connected with the condenser and under vacuum may be piped up in the same manner and cognizance taken of leakage outward only in the computing of the test results.

With suitable precautions, the **weighing or measuring of boiler feed water** should give results accurate to within plus or minus 3 per cent. To secure this accuracy, it is necessary to reduce the effect of the inevitable uncertainties of this method by the exercise of the greatest care, and by the prolongation of each **constant-load run to not less than 10 hours' duration.**

In cases in which it is impossible to arrange for weighing or measuring condensate or boiler feed water, the steam rate of a turbine may be determined by removing the first-stage nozzle block and determining the relation between total flow and initial steam pressure. Several precautions are necessary. The nozzle block must be replaced in such a way that there is no possible leakage of steam around it. For this reason it is recommended that this method never be used if it is necessary to break a high-pressure steam joint within the turbine in order to remove the nozzle block. The determination of steam flow through the nozzles must be made by the actual full flow of steam into a surface condenser, the condensate being weighed or measured. If the first-stage nozzles are in two or more groups, these may, of course, be tested separately. Care must be taken in measuring steam pressures during both the nozzle test and the turbine test, the pressure connections being located at precisely the same points in both tests. The initial pressure must be taken in the chamber from which the nozzles draw steam directly—this is sometimes called the “bowl” pressure. The steam pressure at the exit from the nozzles must not exceed 58 per cent (page 195) of the absolute steam pressure in the chamber from which these nozzles receive steam. The chamber into which the discharge occurs and which connects the nozzle block to the surface condenser must be of ample size to avoid cross currents or eddies, and to give a uniform static pressure throughout the region into which the nozzles discharge. The static pressure must be taken at one or more points in this chamber to be certain that the nozzle pressure therein does not exceed 58 per cent of the initial absolute pressure.

Incidental steam supplied to any auxiliary which is included in the unit being tested, but not included in the main condensate, can usually be determined by a steam meter. In this connection the consideration of relative accuracy is of special importance. For example, suppose a turbine to be served by auxiliaries whose steam consumption is 1 per cent of the steam consumption of the main unit. For every 1,000 pounds of steam used by the main unit, the auxiliary device receives 10, and the two together receive 1,010. The main supply shall be measured with great care—assume that it has been measured with a probable error of not more than 1 in 1,000. This will be an absolute error of 1 in 1,010 of the aggregate steam consumption of the turbine and its auxiliaries. A meter measuring the smaller quantity of steam passing to the auxiliaries need not have a greater absolute accuracy than that attained in measuring the larger quantity, that is, it may have an absolute error of 1 in 1,010 of the aggregate steam consumption. Since this meter measures only 10 in 1,010 of the aggregate steam consumption, its error relative to its own reading may be as great as 1 in 10 without introducing errors in the final result greater than those probable from other sources. Accordingly, relatively small portions of the total steam may be measured by flow meters carefully adjusted and accepted with only approximate calibration or, in extreme cases, without calibration. On the other hand, the performance of auxiliary devices is not without interest, and the steam supplied to them should always be measured with as great accuracy as the circumstances permit. The fact that large relative errors are permissible in such quantities is no justification for failing to take ordinary precautions to minimize all known and controllable errors. The steam escaping from all steam seals shall be eliminated or, if it cannot be reduced to a quantity which is agreed to be negligible, it shall be caught, condensed, and weighed or measured. The steam may be condensed by either an air-cooled or a water-cooled surface condenser or in a bucket of ice water, weighed before and after.

In determining the total steam passing the throttle of a turbine operating on the *regenerative cycle*, several precautions are necessary. There are several ways of disposing of the drips from the extraction heaters, but all methods finally discharge the drips into the main stream of water, leaving the unit to return to the boiler room. Care must be taken to intercept the water stream, for weighing or measuring, beyond

the point at which such drips return, so that the condensate from all steam which passes the throttle shall be included. It will often be the case that the water at this point is too hot to be weighed or measured accurately, because of excessive evaporation. In such cases it will be necessary to install temporarily a device for cooling the water to 140°F. or below.

Condition of Steam Supplied.—The initial condition of steam supplied must be determined in the steam line on the boiler side of the throttle valve¹ and strainer and as close as possible to the throttle valve.

The *pressure* shall be measured by means of a calibrated gage of the Bourdon or dead-weight type, whose instrumental error must not exceed 1 per cent of the pressure indicated. The gage connection shall preferably be in a straight run of pipe at least five diameters long and two and one-half diameters from the discharge end of the pipe.

The *quality* of wet steam shall be determined by means of a steam calorimeter of approved type. It must be borne in mind that the moisture content of wet steam has a disproportionately large effect on the turbine steam rate, and that the quality of steam is very difficult of accurate determination. Accordingly, when a turbine under test is supplied with wet steam, the test results are very likely to be considerably less accurate than those obtained with superheated steam. If the calorimeter indicates moisture in excess of 2 per cent, the steam-rate test cannot be regarded as reliable. Every effort should be made to avoid testing any turbine with a wet-steam supply. Even in turbines designed to run on saturated steam, it is better to test them with steam somewhat superheated, say 25°F., and then, by means of suitable correction factors, to make whatever comparisons are called for.

The *temperature* of superheated steam shall be measured by means of a mercury-in-glass thermometer, having an **engraved stem at least 12 inches long**, accurate to 1°F., which will not break its thread on cooling, and which will return to within 1° of its original reading after quick heating to steam temperature and cooling to room temperature. This thermometer shall be calibrated both before and after the test by one of the methods described on pages 49 to 59.

Should a thermometer be broken during a test, it shall be replaced by a similar instrument, likewise calibrated, and such reserve thermometer shall be provided and calibrated before the beginning of the test. The steam thermometer shall be inserted in a **finned well** of the type described on page 41, **filled with mercury**, fused sodium nitrate, soft solder, or other material. The well shall preferably be located in the middle of a straight run of pipe five diameters long. Stem corrections shall be made

¹ By "throttle valve" is meant the steam stop valve used to start or stop the turbine, which, when open, gives a practically negligible pressure drop. This should be distinguished from governor or regulator valves. Of course, the steam strainer must be clean and cause no abnormal drop in pressure.

as explained on page 45, temperature of emergent stem being taken by means of an auxiliary thermometer.

Condition of Steam Rejected.—For consistent accuracy, exhaust pressure must be measured much more accurately than initial pressure. For condensing turbines, exhaust pressure shall be measured by means of carefully constructed mercury columns, accurate to 0.01 inch of mercury.

Exhaust pressure shall never be computed from temperature observations. Precautions for its measurement are given on pages 387 and 389.

Time.—The exact time (accurate to 1 second) shall be recorded for starting and stopping the weighing or measuring of water for each run, and for the initial and final readings of integrating load instruments for each run. If a run is to be divided into parts for checking, the exact time of the intermediate water check and integrating meter reading shall also be recorded. The approximate time (to the nearest minute) of all other observations shall be recorded. Simultaneous observations may be controlled by the use of whistles, bells, or the like, operated from a common electric circuit, with warning and final signals for each reading. Another convenient method is the use of two or more synchronized stop watches. These are started simultaneously (a little practice makes this easy) a few minutes before the beginning of a test run and are given to the observers of load and water weight, with the understanding that the run will start on the n th minute and stop on the m th minute, with other intermediate points, if desired. The water weighers should record the time, accurate to 1 second, on each occasion when the flow of water is diverted from one tank to another. Such data are of great value in checking for errors in weighing, such as the loss of a tank of water.

Temperatures.—It is usually helpful in interpreting test results to have observations of certain temperatures. When saturated steam is supplied, its temperature serves as a check on the indications of the pressure gage; however, it must be borne in mind that steam may be slightly superheated by certain types of boilers, even though no superheater is installed. In like manner, the *temperature of the exhaust steam* is a rough check on the readings of exhaust pressure. A thermometer for this purpose, however, must be installed with considerable care and good judgment, lest it indicate the *temperature of the metal of the exhaust nozzle* rather than the actual steam temperature. In the *testing of small turbines running on superheated steam, the exhaust is usually highly superheated*, and its temperature is often an important quantity which should be measured with every effort to secure an accurate result.

Preparation for Test.—If the total steam is to be determined by weighing or measuring condensate, the condenser shall be tested for leakage before the test. The **turbine shaft seals** shall be tested for leakage of steam out or air in and shall be adjusted to their normal condition before beginning a test.

If the total steam is to be determined by weighing or measuring boiler feed water, the necessary requirements must be carried out before the

beginning of the test. A preliminary steam-rate or capacity test shall be run for the purpose of determining whether the turbine is in condition suitable for the conduct of a formal test, for the checking of all instruments, and for the training of personnel.

Constancy of Conditions.—During each run of a test, any influence whose variation may affect the results of the test shall be allowed to become as nearly constant as possible before the run begins and must be so maintained throughout the run. The use of *graphic logs* is sometimes of value in determining the proper starting point for a run.

For a turbine running alone, the maintenance of constant load is a special problem in each case. A water rheostat, or special adjustments of load, may be adopted.

In tests of throttle-governed turbines in which manually operated valves are provided to nozzle sections, by means of which better steam consumption may be secured at fractional loads, such valves shall remain open throughout the tests, the turbine thus being able to carry the maximum specified load. This requirement is made because the turbine is not in operating condition if, during the test, it is incapable of swinging up to maximum load at any time. On the other hand, manual valves provided for carrying overloads should be closed during tests under usual operating conditions.

Duration.—Each constant-load run comprised in a steam-rate test should continue for **not less than 1 hour when condensate is weighed or measured**. When boiler feed is weighed or measured, each constant-load run shall continue for **not less than 10 hours**. Each test run shall be preceded by a preliminary period adequate to establishing constant conditions, as shown by suitable quantitative observations. During this preliminary run the load shall be maintained constant and at the same value as during the period during which observations are made.

Starting and Stopping.—A test shall start (and stop) with the simultaneous initiation (or discontinuance) of those power or output readings which are read from integrating instruments, and of water weighing or measuring. It is essential that the observations of these two quantities be started and stopped strictly simultaneously. All other readings shall start somewhat before and stop somewhat after these times. During the period preliminary to a test run, readings shall begin after the load and other operating conditions have been held practically constant for about 15 minutes. Steam rate shall be computed for successive periods of from 3 to 15 minutes (such periods need not be of equal duration), and the formal run shall be deemed to have started with the first two such successive periods for which the uncorrected steam rate checks within 3 per cent. This checking of successive periods should be continued throughout the run, as an indication of the constancy of operating conditions.

Operation at Specified Conditions.—Every effort must be made to run the tests with the rated, or “guarantee,” or other specified values of all conditions, such as pressure, superheat, or vacuum. Previous to the test there should be ready the initial calibrations of all instruments involved in reading initial steam pressure, steam temperature, exhaust pressure, barometric pressure, speed, and power. During the preliminary period, readings of all these quantities are to be taken and the calibration and other corrections are to be **applied immediately**. Whatever adjustment is possible should be made in an effort to **set all conditions at the specified values**. Very often inattention to the barometric pressure, to its correction for temperature, or to a column of water on an exhaust-pressure U-tube or a steam-pressure gage results in running a test with one or more conditions unnecessarily different from the specified values.

In particular, every effort must be made to test the turbine with an exhaust pressure as near as possible to the specified value, because **corrections for variations in exhaust pressure**, especially when extrapolated, are of very **uncertain accuracy**, owing to extreme sensitiveness of steam consumption to steam congestion in the lower-pressure stages, which is greatly influenced by exhaust pressure.

Corrections for Operating Conditions.—After all efforts have been made, it may still be impossible to have every condition as desired, and the test results must be corrected so as to give, as nearly as possible, the values which would have been obtained with specified conditions.

The necessary corrections are given below: Each one gives the percentage change in the steam rate or the total steam for the stated change in the given variable, all other conditions except load remaining constant; that is, all corrections are mutually exclusive.

a. The pressure correction is the percentage change per 10 pounds per square inch. The maximum correction shall not be for more than 2 per cent above or 10 per cent below the specified absolute pressure (page 399). There is also a superheat correction for cases of pressure variation when the superheat also varies.

b. The superheat correction is the percentage change per 10°F. change in superheat (page 399). The maximum correction shall not be for more than 50°F. either way. No test with superheat less than 25°F. shall be run on a turbine designed for superheated steam. For turbines running with high back pressure, the superheat correction becomes very large and is attended with some uncertainty. In such cases the range over which correction may be made must be reduced. In the case of tests with initial superheat of 25°F., or lower, a throttling calorimeter shall be installed and observed, since it is possible to have coexistent liquid water and moderately superheated steam. Should the calorimeter indicate the presence of liquid water, the accuracy of the test as well as the establishment of suitable corrections is open to serious question. Efforts should be directed to improving operating conditions rather than to securing corrections. In tests with moderately superheated steam, all drips should be permitted to blow steam because water and superheated steam may persist together. With drips blowing freely, there is greater certainty that moisture is not reaching the turbine. When the steam rate of a turbine is determined by weighing or measuring the water

fed to a boiler, it will not be possible to permit drips to blow, unless special precautions are taken to determine the quantity of steam and water discharged.

c. Moisture Correction.—Where guarantees are made for wet steam, this correction is the percentage of change in steam rate for each 1 per cent change in initial moisture. The maximum correction shall not be for more than 2 per cent variation of moisture from specified conditions. Tests with saturated steam (wet or dry) are necessarily of a low degree of accuracy and should therefore be avoided if possible. Turbines guaranteed for saturated steam are much more accurately tested with slight superheat and if this is done a superheat correction curve (page 399) is needed.

d. The vacuum correction is the percentage change per inch of mercury change in absolute exhaust pressure. The maximum correction shall be for not more than 1 inch of mercury if the absolute exhaust pressure during the test is higher than that specified, nor more than 0.3 inch of mercury if the absolute exhaust pressure during the test is lower than that specified. Different corrections are required for different loads, as half load, full load, one and one-quarter load, etc. (page 400).

e. The back-pressure correction (for noncondensing turbines) is the percentage change per pound change in back pressure. The maximum correction shall be for not more than 20 per cent of the rated absolute exhaust pressure, nor in any case for a variation of more than 10 pounds per square inch.

f. The speed correction is the percentage change for a 10 per cent change in speed. The maximum correction shall not be more than 5 per cent either way.

g. The power-factor correction is the percentage change for each 10 per cent change of power factor. The maximum correction shall be for not more than 20 per cent difference of power factor either way.

h. Load Curve.—A curve must be provided showing total steam, corrected to specified conditions, plotted as ordinates, and actual load in kilowatts, or the like, as abscissas. This is sometimes known as the "Willans line" (page 370). In case the performance depends on different combinations of hand valves, the curve must have branches from tests for each combination. The difference between values on this curve of total steam at rated load (or other standard load), and of total steam at the corresponding actual test load, divided by total steam at rated load (or other standard load), multiplied by 100, gives in percentage the correction to be used as explained for the other corrections.

Correction Tests.—At least one set of correction tests shall be run with loads near the rated value and reduced so as to give correction curves (page 398) for rated load. Other sets may be required for other loads. This is especially true for the vacuum correction, for which values must always be determined over the complete range of both load and vacuum covered during the steam-rate tests. A correction curve in each case should extend as nearly as possible to the specified value. Extrapolation by extension of a curve may sometimes be necessary, however, and in such cases the curve should extend backward a considerable amount in order to fix the correct shape.

The correction curve for initial steam pressure may be obtained by throttling, with further correction for changed superheat from a superheat-correction curve, or by proper boiler control. The initial pressure-correction curve on a turbine with valves opening at loads slightly above a specified load should be run with such loads as will put the valves, etc., in the position designed for the specified load.

The correction curve for superheat may be made by varying the rating at which the boilers are operating. If several boilers are available, the superheat may possibly be varied by changing the number of boilers supplying steam to the turbine, and hence the rating of these boilers. Separately fired superheaters may be used. Otherwise superheat is practically uncontrollable. The reduction of superheat by spraying water into the steam line usually is not permissible, because of the difficulty of getting

a thorough mixing. Decrease of superheat usually gives but slight decrease of output, so that usually a superheat curve may be run at the specified load. Whenever necessary, however, the load must be decreased so as to put the valves, etc., in the position designed for the specified load.

The exhaust pressure of a condensing turbine can be raised by admitting air into the condenser or suction line of a dry-vacuum pump by means of a valve whose adjustment gives a means of control. However, the exhaust pressure cannot be lowered below the value attained with the maximum flow of the coldest water available, together with the maximum-capacity operation of the air and condensate removal pumps. In case of a noncondensing turbine, the back pressure may be varied in one direction by throttling with an exhaust valve, and in the other direction by the use of a test condenser or by providing a free atmospheric exhaust instead of an exhaust head or by some similar expedient.

An exhaust-pressure, back-pressure, or vacuum curve must be run at the constant specified flow computed from the specified steam rate and load. This curve will, of course, be run with specified initial steam pressure and superheat, so that constant specified flow puts the valves, etc., in the position designed for the specified load.

The load curve is usually made as a part of the regular tests and, if so, no special correction curve is needed. There are usually regular tests at a number of loads, all at conditions as near those specified as possible, and these are to be used in the manner given later (page 392).

Tests for the various correction curves usually have slight variation in all of the conditions in addition to the one for which the correction curve is run. A first approximation must be made by assuming corrections for all of the variables, except the one in question, including the load correction. This gives a *tentative set of correction curves* for rated load as well as the tentative total steam-load curve. From these curves a better set of tentative correction factors is to be obtained and all of the tests worked up again, giving a *second set of correction curves* for the rated load as well as a total steam-load curve. From this set of curves, another set of curves is to be drawn if necessary, until finally a set is obtained which would not be altered by further trials.

The correction tests shall be finally plotted so as to show the test values of steam rate for various values of a variable condition as ordinates, and the variable condition as abscissas. A straight line often represents the points, but sometimes a curve must be drawn. The slope of the straight line or of the **chord of a curve** (the chord being drawn between the test value and the specified value of the variable) divided by the specified steam rate gives the *percentage correction*. Often a curve is also drawn with the loads which existed for each correction test as ordinates and the variable condition as abscissas.

The correction obtained directly from the correction tests reduces conditions to the specified value of the variable, for the load shown by the latter curve at the specified value of the variable, since the curve of load against a variable condition will pass through the point giving the specified load at the specified value of the variable. Exactness in this matter is not essential, however, because percentage-correction curves differ but little for wide variations of load. Hence absolute accuracy in setting the specified valve position is not required for correction tests.

The **general formula** for a correction for A.S.M.E. acceptance tests may be developed as follows:

R_s = steam rate at specified conditions.

R_t = steam rate at test conditions.

V_s = the specified value of one of the variable conditions (for example, pressure, temperature, etc.).

V_t = the value of the same variable prevailing during the steam-rate test,

C = percentage correction per unit for this variable condition.

$$C = \left[\frac{R_t - R_s}{V_t - V_s} \right] \times \frac{100}{R_s}$$

If the correction curve is a straight line, the correction C will be constant for all values of R_t and V_t ; but if the correction curve is not straight, the ratio of the change in R to the change in V must be taken as the slope of the chord joining the points on the correction curve corresponding to the actual values of V_s and V_t .

The total correction is a combination of the separate corrections for the several variable conditions.

Let C_p = percentage correction for the actual deviation of initial pressure¹ of the test p_t from the specified value p_s = $C(p_t - p_s)$

C_t = the same for initial temperature,¹

C_x = the same for exhaust pressure—and so on for all variable conditions.¹

Since each of these is expressed as a percentage of the steam rate at standard conditions, the following relation holds:

$$R_s \left(1 + \frac{C_p}{100} \right) \left(1 + \frac{C_t}{100} \right) \left(1 + \frac{C_x}{100} \right) \dots = R_t$$

This may be solved for the value which the steam rate would have were all conditions standard:

$$R_s = \frac{R_t}{\left(1 + \frac{C_p}{100} \right) \left(1 + \frac{C_t}{100} \right) \left(1 + \frac{C_x}{100} \right) \dots} \quad (90)$$

If the various corrections are small, a close approximation is given by the following equation:

$$R_s = \frac{R_t}{1 + \frac{C_p + C_t + C_x + \dots}{100}} \quad (90a)$$

This last equation, however, is not strictly accurate; wherefore, *in all cases of acceptance tests*, the several corrections shall be combined by multiplication as in Eq. (90), and not by addition.

Selection of Load to Correspond to a Condition below Standard.—In running a regular test, when any one of the conditions is unavoidably below the specified value, or in running auxiliary tests for corrections, with any condition purposely set below the specified value, it is important to select a load correspondingly low in order to avoid operation of the turbine in a region which will introduce irregularities other than that corresponding to the single changed condition.

For instance, if a turbine has hand or automatic valves of any kind designed to open at a load slightly above the rated load, then operation at the rated load when there is a condition below that specified would make the turbine function in a manner which was not contemplated for rated load. Hence, a load always must be set so as to put the valves, etc., in the position designed for the specified load with all conditions as specified. This will give a lighter load, with any condition below the specified value. The correction for variation of any one condition is usually the same for a fairly wide variation of load, so that so far as a correction itself is concerned, it is not important to hold an exact load. For all of these reasons, correction tests should usually be run at constant valve position, and not at constant load. The standard position of the valves need not be set with absolute accuracy, because the correction for difference of load is readily given by the load curves.

¹ Percentage corrections must be used in the following formulas with due regard to algebraic sign.

Use of Corrections.—Each correction may be applied either to the total steam or to the steam rate.

For each correction there are given the maximum limits for which it should be used. Usually all corrections, except the vacuum and speed corrections, apply to any load between 50 and 125 per cent of the rated load, and this is to be assumed unless there are tests or specifications to the contrary. The vacuum and speed corrections do not apply over any material range of load.

In each case percentage corrections are computed on the basis of the performance of the turbine at the conditions specified, which is taken as 100 per cent. This means that the net percentage correction for any item of variation is to be subtracted from or added to unity and the result used as a divisor for the test value of total steam or steam rate to obtain the corrected value. Such a corrected value gives the total steam or steam rate corrected to the specified value of the condition in question, for the specified load. A further correction must be made for load as specified by the last item, if the test load differs from the specified load. The several divisors giving the corrections for each condition which varies are to be multiplied together and used as a single divisor for the test value of total steam or steam rate to obtain the corrected value.

Example.—The following hypothetical example will serve to illustrate the method.

Initial steam pressure is low, making the actual steam rate 2.1 per cent higher than it would have been with specified initial pressure. Correction divisor = 1.021.

Initial superheat is high, making the actual steam rate 2.9 per cent lower than it would have been with specified superheat. Correction divisor = 0.971.

Exhaust pressure is high, making the actual steam rate 1.8 per cent higher than it would have been with specified exhaust pressure. Correction divisor = 1.018.

The load is low, making the actual steam rate 0.4 per cent higher than it would have been with specified load. Correction divisor = 1.004.

The combined correction divisor will be $1.021 \times 0.971 \times 1.018 \times 1.004 = 1.013$.

That is, with all things considered, the actual steam rate is 1.3 per cent higher than it would have been with all conditions conforming to the specifications.

Actual steam rate = $1.013 \times$ steam rate under specified conditions.

Steam rate under specified conditions = $\frac{\text{actual steam rate}}{1.013}$.

Correction of Total Steam or Steam Rate.—The observed total steam or steam rate, for each constant-load test run, shall be corrected as follows: (a) If condenser leakage is measured by a continuous method (chemical or electrical), the observed total steam or steam rate for each constant-load test run shall be reduced by a percentage which is the average result of not less than three leakage determinations conducted during that same constant-load test run. If leakage is determined by either of the direct weighing methods, the observed total steam or steam rate for each constant-load test run shall be reduced by the percentage which is the average result of the leakage tests between which the constant-load run is conducted. (b) For water other than condensate and condenser leakage entering the condenser, and for condensate which does not pass the measuring or weighing tanks. (c) For operating conditions.

If the turbine is tested at several loads covering an appreciable range, the total steam, in pounds per hour, shall be plotted against the average turbine load for the respective runs, and the total steam curve shall be drawn. The curve of steam rate, in pounds per unit output, against load, shall be computed point by point from the curve of total steam against load, and not from the actual observed values. For purposes of comparison, the values of steam rate computed from the observations shall be plotted also.

Heat (B.t.u.) Rate.—For turbines operating on the *regenerative* or *reheating* cycles, the steam rate, in pounds per unit output, becomes of small value for comparative purposes. In such cases it is necessary to compute the heat consumption, in B.t.u. per unit output, as follows:

$$\text{B.t.u. per unit output} = h_{gt} + Q_r - (h_{fw} - h_{fc}) - h_{fe} \quad (91)$$

where h_{gt} = total heat of steam supplied on the boiler side of the throttle valve and strainer, in B.t.u. per unit output.

Q_r = heat added to steam by *reheating*, equal to the increase in heat content from the point at which steam leaves the turbine to be reheated to the point at which the reheated steam reenters the turbine casing, in B.t.u. per unit output.

h_{fw} = heat of feed water leaving the highest temperature heater, in B.t.u. per unit output.

h_{fc} = heat content of condensate (condensed steam) at the temperature actually prevailing in the condenser hot well during the test, in B.t.u. per unit output.

h_{fe} = heat content of water at the temperature of the boiling point corresponding to the absolute pressure prevailing at the turbine exhaust flange, in B.t.u. per unit output.

These quantities, it is to be noted, are expressed in B.t.u. per unit of turbine output; that is, each is calculated by multiplying a steam or water quantity by a heat content or a change in heat content, in B.t.u. per pound.

In the case of the complete expansion turbine with *neither reheating nor regeneration*,

$$Q_r = 0, \text{ and } h_{fw} = h_{fc}, \text{ so that} \\ \text{B.t.u. per unit output} = h_{gt} - h_{fe}. \quad (91a)$$

Steam-cycle Efficiency.—The *efficiency of the ideal turbine* operating under the same conditions as the actual turbine shall be computed.*

Engine Efficiency.—The engine efficiency of the actual turbine tested is the ratio of the steam rate or the heat rate of the ideal turbine to that of the actual turbine. For turbines operating on the *regenerative* or *reheating* cycles, only the heat-rate ratio has meaning, since the steam rate alone does not express the thermal economy of the unit (Item 24, page 396).

Data and Results.—The data and results should preferably be reported in accordance with the form in Table XX.

If steam-rate tests have been run at several loads, the principal results shall be shown in curve form. Such curves shall all be plotted with a common scale of load as abscissas, with the following ordinates: (a) total steam, to as large a scale as possible; (b) steam rate, or heat rate; (c) thermal efficiency; (d) engine efficiency; (e) such stage steam pressures as have been observed.

For a turbine governed by throttling, as long as the overload by-pass valve remains closed, these *curves should be straight lines*, similar to the Willans line, provided the initial steam conditions and exhaust pressure do not change greatly. Any marked departure from this straight line is a strong indication of erratic results and should call for a verification of the points in question by further testing. An added use of this curve is in checking the internal condition of the turbine. The position and slope of this line vary with the mechanical condition of the turbine and often reveal circumstances calling for prompt investigation.

As a check on operating conditions, the report should present for each test run graphic logs, with time as abscissas, and with the following ordinates: (a) initial steam

*Ideal steam engine cycles are explained in detail in Moyer, Calderwood and Potter, "Elements of Engineering Thermodynamics," 5th ed., pp. 117 to 136, John Wiley & Sons, Inc., 1933.

pressure; (b) initial steam temperature; (c) exhaust pressure; (d) load; (e) rate of flow of condensate.

TABLE XX.—DATA AND RESULTS OF STEAM-TURBINE $\left\{ \begin{array}{l} \text{STEAM-RATE} \\ \text{CAPACITY} \end{array} \right\}$ TEST

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

Description, Dimensions, Etc.

7. Type of turbine (impulse, reaction, or combination)
 - (a) Number of stages.....
 - (1) Impulse (note which impulse wheels, if any, have two or more "velocity" stages)
 - (2) Reaction (pairs of rows, one stationary, one moving)
 - (3) Extraction stages
 - (4) Reheating stages
 - (b) Condensing or noncondensing.....
 - (c) Type of governor.....
 - (d) Rated speed..... r.p.m.
8. Turbine rating:
 - (a) Load at best economy hp. or kw.
 - (b) Maximum overload capacity of turbine.....hp. or kw.
9. Class of service (driving of electric generator, centrifugal pump, etc.).....
10. Driven unit: type, size, rating, etc., of driven unit (in particular for electric generator).....
 - (a) Maximum continuous rating of generator.....kw.
 - (b) Power factor.....per cent
 - (c) Overload capacity for...minutes.....kw.
11. Auxiliaries:
 - (a) Condensing equipment.....
 - (1) Type.....
 - (2) Make.....
 - (3) Rated capacity.....
 - (b) Oil pumps.....
 - (1) Type.....
 - (2) Drive (direct or independent).....
 - (c) Auxiliaries to driven unit (include in this item size, type, drive, etc. of exciters, ventilating fans, oil pumps, etc. that are auxiliary to the driven unit)

Results

12. Run number.....
13. Duration.....hr.
14. Average load.....hp. or kw.
15. Initial steam pressure.....lb. per sq. in. abs.
16. Initial steam temperature.....°F.
17. Exhaust pressure.....lb. per sq. in. or in. of mercury at 32°F., abs.
18. Temperature of condensate returned.....°F.
19. Actual steam rate.....lb. per hp.-hr. or lb. per kw.-hr.
20. Steam rate corrected to guarantee conditions....lb. per hp.-hr. or lb. per kw.-hr.
21. Guarantee steam rate.....lb. per hp.-hr. or lb. per kw.-hr.
22. Heat rate, actual B.t.u. per hp.-hr. or B.t.u. per kw.-hr.

23. Thermal efficiency, actual.....per cent
 24. Engine efficiency, actual (p. 392).....per cent

Output

25. Output, horsepower-hour, kilowatt-hour, foot-pound, or the like, if observed independently.....
 26. Average load during run, horsepower or kilowatt. This may be observed directly, Item 25 being omitted.....
 27. Net output of turbine generator.....kw.
 28. Output of generator (repeat for each generator)
 (a) For alternating-current generator.
 (1) Phase 1.....kw.
 (2) Phase 2.....kw.
 (3) Phase 3.....kw.
 (4) Total.....kw.
 (b) For direct-current generator
 (1) Voltage.....volts
 (2) Current.....amp.
 (3) Power.....kw.
 29. Additional data for alternating-current unit
 (a) Voltage.....volts
 (b) Frequency.....cycles per sec.
 (c) Power factor.....per cent
 (d) Excitation (repeat for each generator)
 (1) Power from external source received for excitation
 (x) Voltage.....volts
 (y) Current.....amp.
 (z) Power.....kw
 (2) Power delivered from exciter to external loads
 (x) Voltage.....volts
 (y) Current.....amp.
 (z) Power.....kw.
 (e) Power for ventilation (repeat for each generator).....kw.

Atmospheric Conditions

30. Barometric pressure:
 (a) Inches of mercury at 32°F.....
 (b) Pounds per square inch absolute.....
 31. Room temperature.....°F.

Condition of Steam Supplied, Observed Just Ahead of the Throttle

32. Pressure (see p. 388):
 (a) By gage.....lb. per sq. in.
 (b) Absolute.....lb. per sq. in.
 (c) Corresponding temperature of vaporization.....°F.
 33. Temperature.....°F.
 34. Superheat.....°F.
 35. Moisture.....per cent

Condition of Steam in Turbine Stages

36. Stage *xy*. Pressure:
 (a) By gage.....lb. per sq. in., or in. of mercury at 32°F.
 (b) Absolute.....lb. per sq. in., or in. of mercury at 32°F.
 37. Stage *xy*. Temperature.....°F. or moisture.....per cent

Condition of Steam Rejected in Exhaust

38. Pressure (see p. 387):

- (a) By gage (vacuum or pressure).....lb. per sq. in. or in. mercury at 32°F.
- (b) Absolute (vacuum or pressure).....lb. per sq. in. or in. mercury at 32°F.

Condensate Returned to Boiler Feed System

39. Temperature.....°F.

Speed

40. Speed.....r.p.m.

41. Variation in speed between no load and full load:

- (a) r.p.m.
- (b) per cent of average speed

Thermal Data

42. Steam tables used.....

43. Steam supplied:

- (a) Heat content.....B.t.u. per lb.
- (b) Entropy.....units per lb.

44. Steam leaving to be reheated: heat content.....B.t.u. per lb.

45. Steam returned after reheating: heat content.....B.t.u. per lb.

(Repeat Items 44 and 45 for each stage of reheating)

46. Temperature of boiling point corresponding to exhaust pressure (Item 38b) .. °F.

47. Heat content of water at temperature of Item 46.....B.t.u. per lb.

48. Heat content of water actually returned to boiler feed system (temperature of Item 39).....B.t.u. per lb.

49. Heat supplied per pound of steam B.t.u. per lb.

50. Net work per pound of steam developable in ideal turbine operating under the same conditions as the actual turbine.....B.t.u. per lb.

51. Thermal efficiency of the ideal turbine.....per cent

Corrections for Operating Conditions

52. Specified conditions:

- (a) Initial pressure.....lb. per sq. in. abs.
- (b) Initial superheat or quality.....°F. or per cent
- (c) Exhaust pressure.....lb. per sq. in. abs., or in. mercury at 32°F. abs.
- (d) Speed.....r.p.m.
- (e) Power factor.....per cent
- (f) Load.....hp. or kw.
- (g) Voltage.....volts

53. Corrections from test to specified conditions, as percentages of specified values:

- (a) For initial pressure.....
- (b) For initial superheat or quality.....
- (c) For exhaust pressure.....
- (d) For speed.....
- (e) For power factor.....
- (f) For load.....
- (g) For voltage.....volts
- (h) Total correction for operating conditions.....

Correction for Water Gained and Lost

54. Condenser leakage (state how determined, and give pertinent data).....lb. per hr., or per cent

55. Other water gained or lost (give complete information, including method of determining quantities).....lb. per hr.

Steam Quantities

(State method of determining total steam)

56. Total steam observed during run.....lb.
 57. Total steam per hour.....lb. per hr.
 58. Total steam per hour, corrected for water gained and lost.....lb. per hr.
 59. Total steam per hour, further corrected for operating conditions.....lb. per hr.
 60. Steam rate, actual test.....lb. per hp.-hr. or lb. per kw.-hr.
 61. Steam rate, actual test, corrected for water gained or lost.....lb. per hp.-hr. or lb. per kw.-hr.
 62. Steam rate corrected to specified conditions.....lb. per hp.-hr. or lb. per kw.-hr.

NOTES AND CALCULATIONS

1 to 24. Comprise a brief summary, and may be reported without the subsequent supporting material.

7 to 11. Description of main and auxiliary equipment.

14. Same as Item 26.

15. Same as Item 32(b).

16. Same as Item 33, or, if appropriate, substitute Item 35.

17. Same as Item 38(b).

18. Same as Item 39 for a turbine with regenerative water heating by extracted steam. For a turbine without regenerative heating, Item 18 is taken the same as Item 46.

19. Same as Item 61.

20. Same as Item 62.

21. Taken from specifications.

22. Heat rate. See Eq. (91) or (91a).....B.t.u. per unit output

23. $\frac{2,545 \times 100}{\text{Item 22}}$ or $\frac{3,413 \times 100}{\text{Item 22}}$ (See p. 74.)

24. $\frac{2,545 \times 100}{\text{Item 61} \times \text{Item 50}}$ or $\frac{3,413 \times 100}{\text{Item 61} \times \text{Item 50}} = \text{Item 23} \times \frac{\text{Item 49}}{\text{Item 50}}$.

25. Output may be measured directly by integrating instruments.

26. Load may be measured directly by dynamometer (p. 158).

27. Item 28 - Item 29(d)(1)(z) + Item 29(d)(2)(z) - Item 29(c).

43. Is taken for the pressure Item 32 and the temperature Item 33, or the superheat Item 34, or the quality Item 35.

44 and 45. Are taken for the conditions given in Items 36 and 37 and, like them, are to be repeated for each stage of reheating.

49. Item 43(a) - Item 44 + Item 45 - Item 48. For a turbine operating without regenerative water heating by extracted steam, substitute Item 46 for Item 48.

57. $\frac{\text{Item 56}}{\text{Item 13}}$

58. Item 57 corrected for Items 54 and 55.

59. $\frac{\text{Item 58}}{\text{Item 53}(g)}$

60. $\frac{\text{Item 57}}{\text{Item 26}}$

61. $\frac{\text{Item 58}}{\text{Item 26}}$

62. $\frac{\text{Item 59}}{\text{Item 26}}$

CHAPTER XIV

METHODS OF CORRECTING STEAM-TURBINE AND ENGINE TESTS TO STANDARD CONDITIONS (SHORT METHOD)

Standard Conditions for Turbine and Engine Tests.—If tests of steam turbines and engines could be always made at some **standard vacuum, superheat, and admission pressure**, then turbines and engines of the same size and of the same type could be readily compared, and an engineer could determine without any calculations which of two turbines or engines was more economical for at least these standard conditions. But steam turbines and engines even of the same make are not often designed and operated at any standard conditions, so that a direct comparison of steam consumptions has usually no significance.

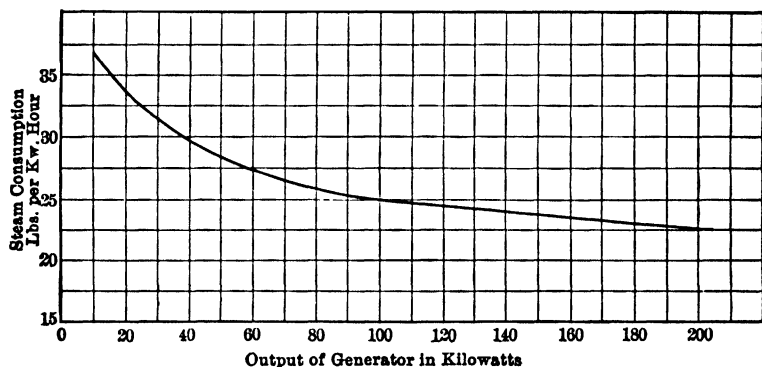


FIG. 274.—Water-rate curve of a typical 125-kilowatt steam turbine. (Generator output.)

It will be shown now how good comparisons of different tests can be made by a little calculation involving the reducing of the results obtained for varying conditions to **assumed standard conditions**. The method given here is that frequently used by manufacturers for comparing different tests on the same turbine or engine (a “checking” process) or on different types to determine the relative performance. To illustrate the method by an application, a comparatively simple test will first be discussed.

Practical Example. Corrections for Full-load Tests.—The curve in Fig. 274 shows the steam consumption for varying loads obtained from tests of a 125-kilowatt **steam turbine** operating at 27.5 inches vacuum, 50°F. superheat, and 175 pounds per square inch absolute admission pressure (at the nozzles). It is desired to find the equivalent steam

consumption at 28 inches vacuum, 0°F. superheat, and 165 pounds per square inch absolute admission pressure for comparison with the "guarantee tests" (Fig. 275) of a steam engine of about the same capacity operating at the latter conditions of vacuum, superheat, and pressure.

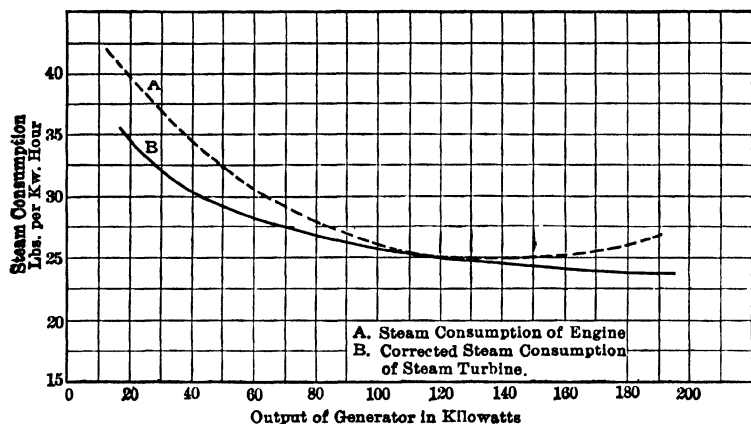


FIG. 275.—Comparative water-rate curves of a reciprocating steam engine and steam turbine. (Both with standard generators.)

The manufacturers of the steam turbine have provided the curves in Figs. 276, 277, and 278, showing the change of economy with varying vacuum, superheat, and pressure. With the help of these **correction curves**, the steam consumption of the turbine can be reduced to the conditions of the engine tests. Figure 276 shows that at between 27 and 28 inches vacuum a difference of 1 inch changes the steam con-

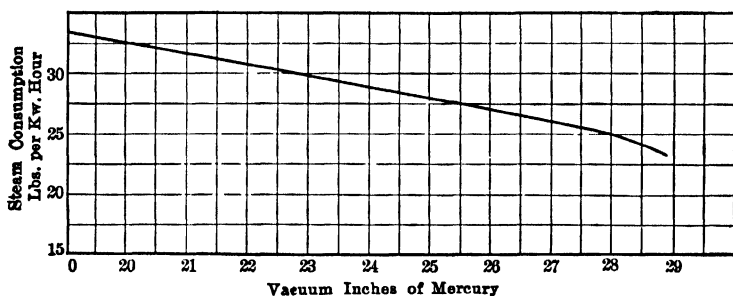


FIG. 276.—Vacuum correction curve for a 125-kilowatt steam turbine.

sumption 1.0 pound. Figure 277 shows a change of 2.0 pounds per 100°F. superheat, and from Fig. 278 we observe a change of 5.0 pounds in the steam consumption for 100 pounds difference in admission pressure. Compared with the engine tests the steam turbine was operated at 0.5 inch lower vacuum, 50°F. higher superheat, and 10 pounds higher pressure. At the conditions of the engine tests, then, the steam con-

sumption of the steam turbine should be reduced 0.5 pound to give the equivalent at 28 inches vacuum but is increased 1.0 pound to correspond to 0°F. superheat, and 0.5 pound more to bring it to 165 pounds absolute admission pressure. The full-load steam consumption for the steam turbine at the conditions required for the comparison is, therefore, $24.5 - 0.5 + 1.0 + 0.5$, or 25.5 pounds.¹

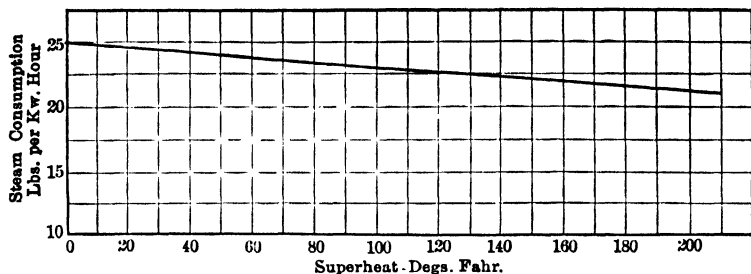


FIG. 277. Superheat correction curve for a 125-kilowatt steam turbine.

Persons who are not very familiar with the method of making these corrections will be likely to make mistakes by not knowing whether a correction is to be added or subtracted. A little thinking before writing down the result should, however, prevent such errors. When the performance at a given vacuum is to be **corrected to a condition of higher vacuum**, the correction must be subtracted, because obviously the steam

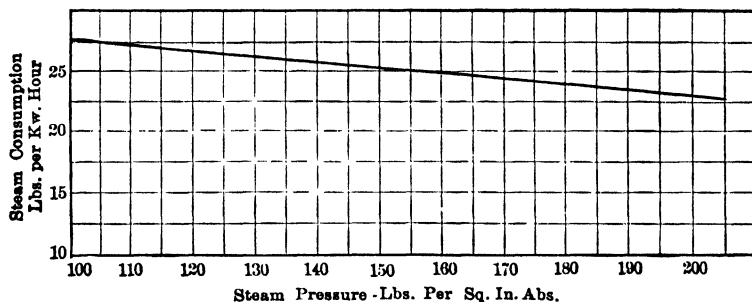


FIG. 278. - Pressure correction curve for a 125-kilowatt steam turbine.

consumption is reduced by operating at a higher vacuum. When the steam consumption with superheated steam is to be determined in its equivalent of dry saturated steam (0° superheat), the correction must be added, because with lower superheat there is less heat energy in the steam and consequently there is a larger consumption. Usual

¹ The corrected steam consumption is found to be nearly the same as that which the three correction curves show for the same conditions, that is, about 25.0 pounds. If there had been a difference of more than about 5 per cent between the corrected steam consumption and that of the correction curves for the same conditions, the "ratio" method as explained on p. 390 for fractional loads should have been used also for full load.

corrections for differences in admission pressure are not large; but it is well established that the economy is improved by increasing the pressure.

Corrections for Fractional Loads by Ratio Method.—It is the general experience of steam-turbine manufacturers that **full-load correction curves**, if used by the following "ratio" or percentage method, can be used for correcting **fractional** or **overloads**. This statement applies at least without appreciable error from half to one and a half load and is the only practicable method for quarter load as well.¹ Stated in a few words, it is assumed then that the steam consumption at fractional loads is changed by the same percentage as at full load for an inch of vacuum, a degree of superheat, or a pound pressure. It will now be shown how this method applies to the correction of the steam consumption of the turbine at fractional loads. Now according to the curve in Fig. 276, the steam consumption at 27.5 inches (25.6 pounds) must obviously be multiplied by the ratio² $\frac{25.0}{25.6}$, of which the numerator is the steam consumption at 28 inches and the denominator at 27.5 inches, to get the equivalent consumption at 28 inches of vacuum. This reasoning

	Conditions of test	Required conditions	Correction ratio	Percentage correction
Vacuum, inches.....	27.5	28	$\frac{25.0}{25.6}$	-2.34 *
Superheat, degrees Fahrenheit.....	50	0	$\frac{25.0}{24.0}$	+4.17
Admission pressure, pounds absolute.	175	165	$\frac{24.8}{24.3}$	+2.06
Net correction.....				+3.89

* Steps in the calculations are omitted in the table, thus $\frac{25.0}{25.6}$ 0.9766 or 97.66 per cent, making the correction 100 - 97.66, or 2.34 per cent. It may seem unreasonable to the reader that these percentages are calculated to three figures when the third figure of the values of steam consumption is doubtful. In practice, however, the ruling of the curve sheets must be much finer and to larger scale so that the curves can be read more accurately.

establishes the proper method for making corrections, that is, that the **base for the percentage** (denominator of the fraction) must be the steam consumption at the condition to which the correction is to be applied.³

¹ A very exhaustive investigation of this has been made by T. Stevens and H. M. Hobart, which is reported in *Engineering*, Mar. 2, 1906.

² Assuming that this short length of the curve may be taken for a straight line without appreciable error.

³ In nearly all books touching this subject so important to the practical, consulting, or sales engineer, the alternative method of taking the steam consumption at the required conditions as the base for the percentage calculation is implied. By such a method percentage correction curves derived from straight lines like Figs. 277 and 278 would be straight lines and, in application, give absurd results. Actually such percentage corrections will fall on curves.

Similarly, the correction ratio to change the consumption at 50°F. superheat to 0°F. is $\frac{25.0}{24.0}$, and to correct 175 pounds pressure to 165 pounds the ratio is $\frac{24.8}{24.3}$. Data and calculated results obtained by this method may then be tabulated as shown in table on page 400.

The signs + and - are used in the percentage column to indicate whether the correction will increase or decrease the steam consumption. "Net correction" is the algebraic sum of the quantities in the last column.

The following table gives the results of applying the above "net correction" to fractional loads.

	One-fourth load 31.3 kilowatts	One-half load 62.5 kilowatts	Three-fourth load 93.8 kilowatts	Four-fourth load 125 kilowatts	Five-fourth load 156.3 kilowatts
Steam consumption from test (Fig. 274).....	31.2	26.9	25.2	24.5	23.6
Net correction + 3.89 per cent.	+1.2	+1.1	+1.0	+1.0	+0.9
Corrected steam consumption.....	32.4	28.0	26.2	25.5	24.5

Curve *B* in Fig. 275 shows the corrected curve of steam consumption for the steam turbine as plotted from the foregoing table. By thus combining, on the same curve sheet, curves *A* and *B* as in this figure, the points of better economy of the turbine are readily understood.

Results of economy tests of various turbines are of very little value for comparison when the steam consumptions or "water rates" are given for all sorts of conditions. With the assistance, however, of curves like those shown in Figs. 276, 277, and 278, if they are representative of the type and size of turbine tested, it is possible to make valuable comparisons between two or more different turbines.

The foregoing method is based on the usual practice of combining the correction percentages in addition as in Eq. (90*a*) (page 390). The more accurate method, of course, is to combine the correction percentages by multiplication as in Eq. (90). In this case, since the net connection is small, the result is about the same by both methods.

CHAPTER XV

TESTS OF COMPLETE STEAM POWER PLANTS—COAL FIRED

These rules are intended to apply to commercial tests of a complete plant to determine the number of pounds of coal consumed per unit of work done in a unit of time. For tests of the component parts of a complete plant, such as boilers, engines, turbines, etc., rules may be found in the respective Codes on the preceding pages applying to such cases.

Read the general instructions given on pages 289 to 303. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

Fuel.—Determine the character of the fuel to be used according to the object in view. For further particulars reference may be made to the Test Code for Boilers and Other Steam Generating Units (page 304).

Apparatus and Instruments.—The apparatus and instruments required for a simple performance test of a steam plant are:

- a. Platform scales for weighing coal and ashes.
- b. Coal calorimeter.
- c. Steam engine indicators.
- d. A speed-measuring apparatus.
- e. Electrical instruments for determining the output of an electric plant.

If the test involves the determination of boiler performance, and engine or turbine performances, additional instruments should be used as pointed out in the respective Codes referring to such tests.

Duration.—The duration of a plant test should be not less than one day of 24 hours, and preferably a full week of seven days, including Sunday.

In cases where the engine or turbine is in operation only a part of the day, the duration on which the results are computed should be considered the length of time that the engine or turbine is in operation at its working speed.

Starting and Stopping.—In a plant operating continuously, day and night, the times fixed for starting and stopping should follow the regular periods of cleaning the fires. The fires should be quickly cleaned and then burned low, say to a thickness of 4 inches. When this condition is reached, the time should be noted as the starting time, and the thickness of each coal bed observed, as also the water levels and the steam pressure.

Fresh coal should then be fired from that weighed for the test, the ashpits thoroughly cleaned, and the regular work of the test proceeded with. At the close of the test, following a regular cleaning, the fires should again be burned low; and when their condition has become the same as that observed at the beginning, the water levels and steam pressure also being the same, the time is observed and this time taken as the stopping time. If the water levels and steam pressure are not the same as at the beginning, a suitable correction should be made by computation. The ashes and refuse are then hauled from the ashpits.

In a plant running only a part of the day, when during the remainder of the day the fires are banked, the time selected for the beginning and end of the test should be that following the close of the day's run, when the fires have been burned low preparatory to cleaning and banking. The amount of live coal left on the grates under these circumstances is estimated at the beginning of the test, and the fires brought to the same condition, as near as may be, at the close of the test the next day. If the two quantities differ, a suitable correction is made in the weight of coal fired, as found by calculation.

Records.—The general data should be recorded as pointed out on page 359, under the head of Records. Half-hourly readings of the various instruments concerned are usually sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals *if the nature of the test makes it necessary*. Mark on each card the cylinder and the end on which it was taken, also the time of the day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

Sampling and Drying Coal.—During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture.

Ashes and Refuse.—The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed in a dry state, and, if desired, a representative sample should be obtained for proximate analysis and the determination of the amount of unburned carbon which it contains.

TABLE XXI.—DATA AND RESULTS OF COMPLETE STEAM POWER-PLANT TEST

1. Test of plant located at
to determine conducted by
2. Type of engine or turbine and class of service
3. Rated power of engine or turbine
4. Type of boilers
5. Kind and type of auxiliaries (air pump, circulating pump and feed pump; jackets, heaters, etc.)

6. Dimensions of engine or turbine.....
7. Dimensions of boilers.....
8. Dimensions of auxiliaries.....
9. Dimensions of condenser.....
10. Date.....
11. Duration.....hr.
12. Length of time engine or turbine was in motion with throttle open.....hr.
13. Length of time engine or turbine was running at normal speed.....hr.
14. Kind of coal.....
15. Size of coal.....

Average Pressures and Temperatures

16. Steam pressure at boiler by gage.....lb. per sq. in.
17. Steam pipe pressure near throttle, by gage.....lb. per sq. in.
18. Barometric pressure of atmosphere, in in. of mercury.....
19. Pressure in receiver by gage.....lb. per sq. in.
20. Vacuum in condenser.....in.
- 20a. Temperature of steam near throttle.....°F.
21. Number of degrees of superheating, if any, near throttle.....°F.
22. Temperature of feed water entering boilers.....°F.

Total Quantities, Time, Etc.

23. Total coal as fired¹.....lb.
24. Moisture in coal.....per cent
25. Total dry coal consumed.....lb.
26. Ash and refuse.....lb.
27. Percentage of ash and refuse to dry coal.....per cent
28. Calorific value by calorimeter test per lb. of dry coal.....B.t.u.
29. Cost of coal per ton of....lb.....dollars

Hourly Quantities

30. Dry coal consumed per hour, based on duration of running period.....lb.

Indicator Diagrams

31. Mean effective pressure, in lb. per sq. in.....

Alternating-current Electrical Data

32. Total electrical output.....kw.-hr.
33. Electrical output per hour.....kw.
34. Output consumed by exciter.....kw.
35. Net electrical output per hour.....kw.
36. Average volts each phase.....volts
37. Average amperes each phase.....amp.
38. Power factor.....

Speed

39. Revolutions per minute.....

Power

40. Indicated horsepower developed by main engine:
 First cylinder.....i.hp.

¹ Where an independent superheater is used, this includes coal burned in the superheater.

Second cylinder.....	i.hp.
Whole engine.....	i.hp.
41. Net electrical horsepower.....	e.hp.

Economy Results

42. Dry coal consumed per indicated horsepower per hour.....	lb.
43. Dry coal consumed per kilowatt-hour.....	lb.
44. Cost of coal per indicated horsepower per hour.....	cents
45. Cost of coal per kilowatt-hour.....	cents

"Plant" as used in this report should include the entire equipment of the steam plant producing power, that is, the main cylinder or cylinders, the steam jackets and reheaters, the air, circulating and boiler-feed pumps if steam driven, and any other machinery driven by steam required for the operation of the engine. That the engine plant should be charged with the steam used by all the auxiliary machinery in determining the plant economy is necessary because the steam consumption of the engine is finally benefited, or at least it should be, by the heat they return to the system. It is, of course, now the general practice in commercially operated plants to pass the exhaust steam from auxiliaries operated non-condensing through a feed-water heater, thus carrying back to the boiler a great deal of the heat.

When a plant is **operating noncondensing**, discharging the steam into the atmosphere, or with a jet condenser, the **steam consumption of the engine cannot be determined by weighing or measuring the steam used** as can be done when a surface condenser is used. The method followed in this case is to determine the steam used by the weight of water supplied to the boiler (page 307), it being assumed that all the steam from the boiler or boilers used goes to the engine tested. It can usually be arranged for a test of one of the engines in a large plant that one or more boilers can be segregated or cut off in the piping connections so that these boilers alone supply the engine. When, however, this method is to be used, it is necessary to determine by a separate test the **leakage of the boiler and of the piping** from the boiler to the throttle valve of the engine. This leakage is, of course, the amount of feed water pumped into the boiler to keep the level in the water gage constant without taking away any more steam than is lost in this way. When this leakage is being determined, the pressure in the boiler must be maintained the same as that at which steam is to be supplied to the engine for its test. The feed water pumped into the boiler supplying the engine less the boiler and pipe leakage will be the net amount of steam used by the engine.

CHAPTER XVI

GAS- AND OIL-ENGINE AND PRODUCER TESTING

The testing of internal-combustion engines of the reciprocating type operating with gas, gasoline, kerosene, and alcohol does not differ essentially in the important details from steam-engine practice already explained in Chap. XII. Indicator diagrams can be utilized to show the inner workings in the engine cylinder, giving a record of the pressure, "timing" of the valves, and ignition for the operation of the engine through a cycle.¹

Certain important precautions should be observed when taking indicator diagrams, so that a reasonable degree of accuracy may be expected from the results of tests. In the first place, the connections between the indicator and the combustion chamber should be as short as possible. It is much more important that in an internal-combustion engine the volume by which the combustion chamber (clearance) is increased by the indicator connections should be small in comparison with the volume of the engine cylinder than in a steam engine, because by increasing the clearance volume, obviously, the pressure resulting from compression is reduced as well as the pressure due to the explosion. It is in this way that large indicators and indicator connections may cause a considerable reduction in the thermal efficiency of an engine, that is, reducing the efficiency of the transformation of heat energy into work.

Tests of gas or oil engines as made commercially have usually three objects in view:

1. Brake horsepower.
2. Indicated horsepower.
3. Gas or oil consumption per horsepower per hour.

Many types of gasoline engines, particularly those designed for the automobile and airplane services, operate at such high speeds that the indicated horsepower cannot be obtained with accuracy. In many other engines in this group the mechanical efficiency is very low. It is for these reasons that such engines are rated by the useful or brake horsepower instead of by the indicated horsepower, as with steam engines.

¹ In what is called usually a four-cycle engine there are four piston strokes—one each for suction, compression, expansion, and exhaust—corresponding to two revolutions of the crank shaft for a complete cycle, while in a so-called two-cycle engine, two strokes of the piston make a complete cycle—corresponding to one revolution of the crank shaft. In the latter case suction and compression are combined in one stroke and expansion and exhaust in another.

Brake horsepower is therefore the prime criterion by which the performance of these engines is expressed.

Ordinary types of steam-engine indicators, moreover, are not very satisfactory for testing slow-speed gas and oil engines, and engineers prefer to use one of the type shown in the accompanying illustration (Fig. 279). It differs from steam-engine indicators of the same type in having in the lower part of the main "barrel" a cylinder of smaller diameter than the one just about it, containing the spring. This smaller cylinder takes a piston of only half the area of the standard size. By this means the scale

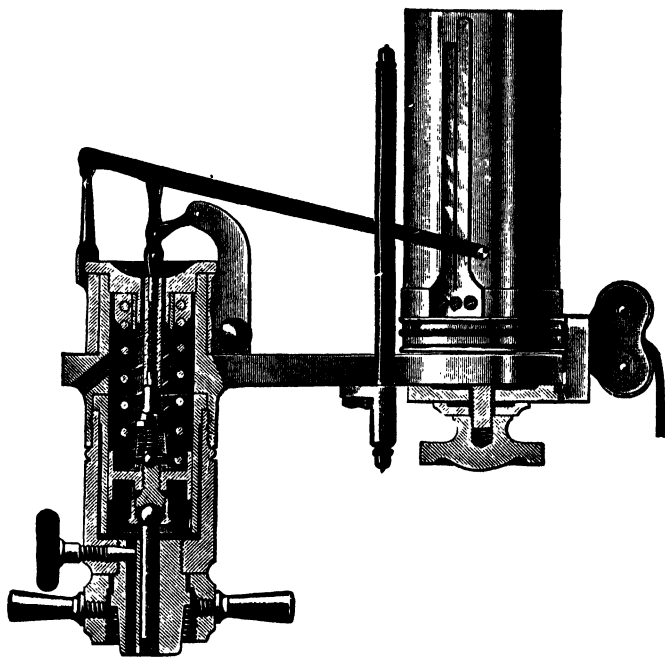


FIG. 279.—Gas-engine indicator.

of the spring ordinarily used is doubled and the shock on the small rods and levers of the pencil motion is only half as great, thus making the liability to breakage and the cost of repairs for such indicators very much less than when a piston of the standard size is used.

Types of indicators for high-speed automobile and airplane engines are explained on pages 125 to 129.

Measurement of the Fuel Used.—For tests of gas engines the gas used is usually measured by means of a gas meter. A so-called "wet" meter (page 182) is always to be preferred; but if a carefully calibrated dry meter is used, fairly good results can be obtained and it is accurate enough for some commercial tests. For gasoline, kerosene, and other oil engines the amount of fuel used is preferably determined by direct weighing. The

author has found the automatic indicating scales of the **pendulum** type¹ now generally used by grocers and meat dealers to be most satisfactory. By this means the weight of the oil remaining in the "supply" vessel can be observed regularly throughout a test just as with a gas meter, so that irregularities in operation can be immediately observed. The vessel containing the oil used by the engine when placed on a scales must be connected to the carbureter or the pump, as the case may be, with a very flexible metallic tube made without rubber insertion.² If an

indicating scales is not available, a very small platform or grocer's beam scales can be used satisfactorily, although the weight of liquid fuel is not easily obtained at regular intervals.

Another method very commonly applied, however, is to use a cylindrical vessel of small diameter provided with a gage glass in which the level of the liquid can be observed. Such a vessel can be calibrated to determine the weight or volume of oil per inch of height measured on the gage glass. An improved device of this kind is shown in Fig. 235 (page 250).

Observations taken for a test of a gas or oil engine are, in general, much more uniform than the corresponding data taken in a steam-engine trial. For this reason gas-engine tests, in particular, can be made of much shorter duration than upon a steam engine for the same degree of accuracy.

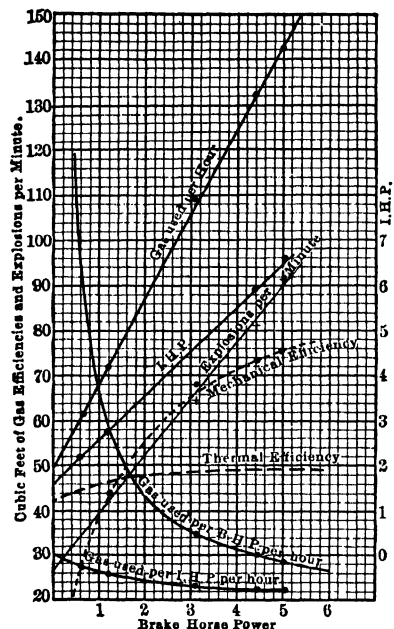


FIG. 280.—Typical economy, speed, horsepower, and efficiency curves of a 5-horsepower gas engine.

For both gas and oil engines the points plotted on a scale of brake horsepower for abscissas and fuel used per unit of time for ordinates will fall along a straight line, similar to and resembling the **Willans** line for steam engines and steam turbines (see page 372). A typical set of curves of the results of a test of a gas engine is shown in Fig. 280. Brake horsepower is taken for the abscissas, as should always be done for gas- and

¹ Very satisfactory scales for this purpose are made by the Toledo Scale Co., of Toledo, Ohio. Similar scales of the **spring** type are not recommended, because necessarily some of the vibrations of the engine will be transmitted to the scales and the indications of the pointer will not be so accurate as they should be.

² Tubes of this kind are made by the Pennsylvania Flexible Metallic Tubing Co., Philadelphia.

oil-engine tests, and gas used per hour, to its corresponding scale of ordinates, is given by the top curve. Other curves show the gas used per brake horsepower per hour and per indicated horsepower per hour, the indicated horsepower, the **mechanical efficiency**, the **number of explosions per minute**, the **revolutions per minute**, and the **thermal efficiency** (heat equivalent of the brake horsepower¹ divided by heat supplied).

If the engine uses gas generated from coal in a producer, the test should cover a long enough period to determine with accuracy the coal used in the **gas producer**; such a test (page 421) should be of at least 24 hours' duration, and in **most cases it should preferably extend over several days**.

Gas bags should be placed between the meter and the engine to diminish the variations of pressure, and these should be of a size proportionate to the quantity used. Where a meter is employed to measure the air used by an engine, a receiver with a flexible diaphragm should be placed between the engine and the meter. The temperature and pressure of the gas should be measured, as also the barometric pressure.

A.S.M.E. TEST CODE (1929) FOR INTERNAL-COMBUSTION ENGINES (ABRIDGED)

Tests for determining the performance of internal-combustion engines by this Code apply to all forms of internal-combustion engines but are limited to the engines alone. Separately driven auxiliaries that are essential to the operation of the engine, such as scavenging pumps or blowers and injection air compressors, for example, may be included in the tests, but the testing of such units must follow the other codes appropriate to the unit to be tested and the results in horsepower absorbed introduced and used in this Code.

Sizes of internal-combustion engines coming within this Code range from one to several thousand horsepower. Speeds range from less than 100 revolutions per minute to over 3,000 revolutions per minute. Numbers of cylinders range from one to 12 or more; some horizontal and others vertical; some single acting and others double acting; some two cycle and others four cycle, with every variety of attached and separate scavenging pump for two-cycle engines. Fuels range from the lightest distillates to the heavy-residue fuel oils, organic and mineral in the liquid-fuel class, and in the gaseous form include every combustible gas commercially available. Stationary engines operate at all loads normally at constant speed under governor control, for any period of time and any degree of intermittence of use. Marine engines operate normally at full load, and the constant speed fixed by the propeller, but must be maneu-

¹ Most engineers consider the thermal efficiency calculated on the basis of brake horsepower more important than that based on indicated power because, particularly for high-speed engines, indicators are not very reliable.

vered ahead and astern at any speed under hand control and may operate for any period of time from short intermittent runs to the continuous full load for a month or more required of seagoing motorships. **Aircraft engines** must meet about the same service conditions as marine engines, except that a smaller length of single run is required, but they must operate in the greatest known ranges of atmospheric pressure and of temperature. Other engines of the **automotive group**, those for land transportation, including those for automobiles, motor trucks, tractors, railroad locomotives, and self-contained cars, are essentially variable in speed as normally used, and equally variable in torque, horsepower being as much a matter of secondary importance as in a hoisting engine or steam locomotive. The engines of this last group drive their vehicles through gear sets of different speed ratios, and tests of them must be conducted with due regard to these service conditions.

Included within this Code are the following internal-combustion engines: (a) stationary engines and electric-generating sets, even if not intended for stationary use, burning gaseous or liquid fuels, including the scavenging pumps of two-cycle engines, and injection air compressors, if used, and separately driven, (b) marine engines, exclusive of propeller, and burning light or heavy liquid fuels, (c) aircraft engines limited to block tests, exclusive of propeller, and burning light and heavy liquid fuels, (d) land-vehicle engines limited to block tests and burning light and heavy fuels.

This Code is limited to tests in which the object is of a commercial nature. It excludes all tests forming part of experimental investigations, research, or development. But while no engineer undertaking tests of a research or development nature needs any Code if he is competent to make such investigations, it will usually be found that parts of such research work will be aided and in no way hampered by following parts of this Code. It is also in the interests of those branches of engineering, whose progress is based on experimental data, if any parts of an investigation are carried out in accordance with the Code, and recorded in accordance with the Code. Results so obtained are more easily comparable with the large numbers of tests always being carried out on the lines of the Code. The research data then enter readily into the general body of knowledge of the internal-combustion engine.

Measurements.—The measurements that must be made in a performance test of an internal-combustion engine will be of the following quantities: (a) cylinder diameter, stroke, and clearance volume of working cylinders; (b) diameters of piston-rods and tail rods of working cylinders of double-acting engines; (c) the brake-horsepower or shaft-horsepower output; (d) the kilowatt output if engine is direct-connected to a generator; (e) the speed in revolutions per minute; (f) the horsepower to drive the water-cooling pump and fuel pump (if any); (g) the horsepower to drive the independent scavenging pump or blower; (h) the horsepower to drive the independent injection air compressor; (i) the amount of fuel supplied: cubic feet of gas for gas-burning engines, or pounds of liquid fuel for liquid-fuel engines; if more than one kind at the same time, the amount of each kind; (j) the calorific value of the fuel, high heating value.

The measurements which may be made in addition to those required above will be some or all of the following, as may be agreed in advance by the test engineers: (a) cylinder diameters and strokes of injection air compressor; (b) two-cycle engine scavenging pump diameter and stroke; (c) diameters of piston and tail rods of scavenging pumps of two-cycle engines; (d) diameters of piston rods and tail rods of injection air compressor; (e) the indicated horsepower; (f) the number of explosions per minute or fuel injections per minute, or, in general, combustion times per minute, for each and for all working cylinders; (g) the spray-air pressure of air-injection oil engines; (h) the exhaust back pressure at the engine; (i) the scavenging-air pressure of two-cycle engines, constant or maximum values; (j) the manifold vacuum of carburetor engines; (k) the jacket-water or oil-supply pressure at the main and, if jackets are divided with separate supplies, the pressure at the supply point or branch pipe to each; (l) the jacket water or oil back pressure at the main outlet or, if jackets are divided with separate discharges, the pressure at the outlet of each; (m) the pressure or vacuum or the hydraulic head, positive or negative of the supply of liquid fuel at the carburetor connection of carburetor engines, and at the injection-pump suction of injection oil engines; (n) the pressure or vacuum of the gas supplied to gas-burning engines at the mixing valve connection; (o) the total pressure, static, and velocity of cooling-air supply of air-cooled engines; (p) barometric pressure; (q) atmospheric humidity; (r) the pressure of lubricating-oil supply to bearings of engines lubricated by pump forced-feed circulation; (s) the temperature of the atmospheric-air supply; (t) the temperature of the gas for fuel at the mixing valve, or for liquid fuel at the carburetor or injection-pump suction; (u) the temperature of the jacket water or oil as supplied to the engine as a whole or, if jackets are divided with separate supplies, the temperature of each separate supply; (v) the temperature of the jacket water or oil leaving the engine as a whole or, if jackets are divided with separate outlets, the temperature at each outlet; (w) the temperature of the scavenging air of two-cycle engines; (x) the temperature of injection air; (y) the temperature of lubricating oil supplied to the engine at as many points as there are different supplies of pump-circulated lubricating oil, or in the crank case of engines without pumps or having pump deliveries that do not permit of temperature measurement; (z) the temperature of the cooling air supplied to an air-cooled engine; (aa) the amount of lubricant consumed, in pounds; (bb) the amount of jacket water or oil supplied, in pounds, for engines as a whole, and when jackets are divided for each if both supply and discharge are separate; (cc) amount, in pounds or cubic feet, of cooling air supplied to air-cooled engines; (dd) amount, in pounds, of jacket water evaporated in jackets of hopper-cooled engines, (ee) amount, in pounds, of injection water supplied internally to cylinders of some oil engines, as a liquid or as steam; (ff) the compression pressure in cylinders when hot and cold, at normal or at reduced speed at wide-open throttle.

Instruments and apparatus required for a performance test of an internal-combustion engine as prescribed by this Code will include some or all of the following: (a) scales with or without special auxiliary apparatus for weighing liquid fuel, with or without tanks to be used in cooperation; (b) gas meters for measuring gaseous fuel, or gas-metering methods with indirect observation apparatus; (c) gas calorimeters for determining the heating value of gaseous fuels; (d) Baumé oil hydrometer for petroleum liquid fuels, and for indirect determination of calorific power; (e) pressure gages and mercury or water columns for measuring small pressures and vacua; (f) thermometers; (g) barometers; (h) gas-engine and oil-engine indicators for working cylinders and steam-engine indicators or special low-scale springs for cylinders of two-cycle scavenging pumps and low-pressure cylinders of injection air compressors; (i) pressure-indicating or recording instruments for compression pressures or injection pressures in cylinders; (j) planimeters; (k) tachometers, revolution counters, or other apparatus for measuring speed or number of fuel admissions per minute; (l) absorption dyna-

mometers of appropriate type for the horsepower, speed, and torque of the engine to be tested, especially hydraulic brakes and electric dynamometers; (*m*) appropriate electrical instruments and apparatus, if the engine is direct-connected to a generator, to provide a suitable electrical load and to measure it.

Use of Indicators.—Indicators cannot be used at all on some internal-combustion engines and in other cases only within limits. In no case can an indicator be regarded as a precision instrument on an internal-combustion engine to the same degree as on a reciprocating steam engine. Some special cases are noted:

a. When the engine speed exceeds 400 revolutions per minute, an indicator must not be used for determining the cylinder mean effective pressure of an internal-combustion engine.

b. When the engine speed exceeds 400 revolutions per minute, suitable indicators may be useful as a means of adjustment of valve or spark timing but have no other value and are of no use even for such adjustments in the case of multicylinder engines.

c. On all hit-and-miss-governed engines the indicator must not be used for determination of the cylinder mean effective pressure for indicated-horsepower determinations.

d. When indicator cards taken even at speeds of 400 revolutions per minute or less are not constant in shape and area over long periods of time while a constant load is maintained, the indicator must be rejected as a means of determining cylinder mean effective pressures though it may be used as a means of adjustment of timing, especially as a guide to equalizing the timing for multicylinder engines or the two ends of a double-acting cylinder. Constancy of area shall be plus or minus 2 per cent, so that, when the largest area differs from the smallest area by 4 per cent or more of their arithmetical mean, all shall be rejected. At various odd intervals of time, at least 10 diagrams shall be taken on one card in addition to those taken at regular intervals, for comparison as to constancy.

e. When indicator cards taken even at speeds of 400 revolutions per minute or less are apparently constant in area, according to the conditions *d*, they may still be useless as a means of determining cylinder mean effective pressure and indicated horsepower, when the cylinder pressures are very high, the spring scale large, 500 pounds more or less, and the whole measured card area at the same time is very small. Under these conditions the precision will be very low and the indicated horsepower much in error. This condition makes it hopeless to undertake determinations of indicated horsepower for many oil engines of the high-pressure type, developing only small mean effective pressures.

f. Even in the cases where the indicator cannot properly be used as an instrument for measuring mean cylinder pressures and indicated horsepower, it can be usefully employed as a means of adjusting and equalizing the cylinder compression, the adjustment of main valves, the point of ignition in electrically ignited engines, of spray-valve timing of Diesel oil engines, or the injection-pump timing of injection oil engines with direct pump injection.

NOTE.—No indicated-horsepower determination should be made unless a separate indicator is used on each working cylinder of a multicylinder engine (each end).

Dynamometers.—Brakes or absorption dynamometers when used must be suitable for the engine to be tested, in capacity, in speed-torque relations, and in ability to maintain an adjustment without the necessity for constant hand manipulation to hold the scale beam steady.

a. For testing engines of the aircraft or land-vehicle types on block tests, nothing but an electric dynamometer shall be used to determine brake horsepower.

b. For brake-horsepower tests of marine engines, the electric dynamometer shall be used, or hydraulic, if "reversing" tests are to be run.

c. Other forms of brake may be properly used when testing stationary engines as well as also, when available, hydraulic and electric dynamometers.

d. When brakes of any kind cannot be mounted directly on the engine shaft without other supports, due precautions must be taken to avoid errors due to these supports.

e. In no case may a brake driven by a belt or otherwise than by direct coupling to or mounting on the engine shaft be used for engine brake-horsepower measurements.

f. With due precautions and under proper provisions brake-horsepower determinations may be made by the indirect or substitution method, and this is important because in some cases it is the only means available. According to this method the engine may be tested with a load, such as a propeller (air or water type), or a fan, which propeller or fan may later be, or previously have been, tested for its speed-torque characteristic curve, and the horsepower of the engine determined by reading from this curve the horsepower for the speed maintained by the engine. The special precautions referred to relate to assurances of identity of conditions of air flow through and resistance of the propeller or the fan under the two separate tests of use and of calibration. This is so difficult that none but test engineers of special experience should undertake it (page 165).

Calorimeters.—The heating value of fuels to be used in calculations of results of tests of internal-combustion engines must be the "high value" with products condensed, or its equivalent. Accordingly the following recommendations are made:

a. When gaseous fuel is burned in an engine under test, a standard form of gas calorimeter shall be used as one of the instruments of the test, and its determination shall be used as the heating value of the gas as supplied. *Calculations of heating value* from the gas analysis are not permissible according to the Code.

b. When liquid fuel is burned in an engine under test, its heating value shall be determined in a standard bomb calorimeter by a recognized physical or chemical laboratory and in the report the "high value" should be used in calculating final results.

c. The heating value of gasolines and kerosenes may be estimated by the Sherman and Kropff formulas, modified by Strong, from the Baumé gravity reading without the bomb test, if such procedure is made the subject of previous agreement.

The Sherman and Kropff formulas for the heats of combustion (B.t.u. per pound) of gasoline and kerosene as modified by Strong (U. S. Bureau of Mines, *Bulletin* 43, page 19) are as follows (B = degree Baumé):

$$\text{Gasoline: high value, } 18,320 + 40(B - 10). \quad (92a)$$

$$\text{Kerosene: high value, } 18,440 + 40(B - 10). \quad (92b)$$

Volume Meters, Gas.—Gas measurements for an engine burning gaseous fuel must be made with meters or by methods yielding the greatest possible accuracy consistent with availability and cost in the large sizes. In no case can the results of proportional meters be regarded as satisfactory, nor those of vane meters of the anemometer type. Within the limits of size and cost, positive displacement meters are to be preferred, but these are not available for measurements of large capacity.

Large-capacity measurements are best made by means of one of the various types of flow-rate measurements, and that one, applied to mixed gases, should be used that requires the simplest and most accurate determination of physical or chemical properties.

If measurements of volume of injection air or of scavenging air are desired, the orifice-flow method (page 190) will be sufficiently correct.

Liquid Measurements.—Liquid fuel supplied to an engine should never be measured by volume; it must be weighed directly as used. Combinations of weighing and volume measurements are acceptable if the volume of the tank itself is not the measure but only a reservoir from which the engine draws a supply, starting with a predetermined level marked by a hook gage or on a glass, and a weighed quantity subsequently supplied to restore the original level. In this case the weight is the measure and the result acceptable if proper precautions are used to limit the weight error of reading levels to something less than the error of the scale weighing. In no case shall meters be used to measure liquid fuel supplied to an engine.

When volatile fuels are used, and especially when heavy liquid fuels containing light volatiles must be heated to make them flow, precautions must be taken to prevent losses of weighed fuel by evaporation.

Leakage Tests.—Leakages must be located and corrected. Among important leakages to be investigated attention is called to the following:

a. Cylinder leakage outward, lowering the compression and efficiency or possibly preventing operation. These leakages may be best checked by an indicator if the engine is one suitable for indicator use. When there is any clear space between the compression and expansion lines of a diagram taken when there is no combustion and when cooling water is not running, it may be assumed that leakage is excessive. In this case, and in cases where an indicator cannot be used to check leakage, the several possible sources of such leakage must be separately checked. These are piston ring, cylinder, cylinder-head gasket, air starting valve, relief valve, spray valve or spark-plug seat, inlet or exhaust valves of four-cycle engines, or air-scavenging valves of valve-scavenging two-cycle engines.

b. Leakage into the cylinder may occur from the jackets through bad joints, cracks or porous spots and blowholes in the castings, compressed air from the starting air system through leaky air starting valves, fuel oil past spray valve at improper times, and high-pressure spray air when air spraying is used in oil engines.

c. Miscellaneous leakages not into or out of the cylinder but elsewhere may be sought; in the manifold system of carburetor engines anywhere between carburetor and inlet-valve seat, especially at flanges and valve-stem guides; fuel oil in injection oil engines at pump valves, pump plunger, oil-delivery pipe, or spray valve.

Operating conditions to be maintained during the test are the ones most uncertain, as will be clear from the following note:

Aircraft engines must normally work in altitudes varying from sea level to perhaps 30,000 feet, and in temperatures varying from highest in summer in the tropics, perhaps 120°F., to the coldest known, under 50°F. below zero, and in any weather—dry, fog clouds, rain, and snow. They must also operate well at any angle of inclination and even upside down, but always with an air-propeller load. Since the only tests contemplated under this Code are fixed block tests, it is clear that service con-

ditions cannot be closely reproduced and only approximated even in the special altitude room of the Bureau of Standards with its air-refrigerating apparatus.

Effect of Load and Speed.—Operating conditions during the test include mainly matters of load and speed to be maintained, with other points incidental to these. These should be agreed upon in all details and should not be inconsistent with the conditions prescribed below:

a. Engines of the stationary class, including all engine-driven generator sets, shall be tested at constant speed under governor control or as near constant as the governor will maintain it, and at whatever load in horsepower or kilowatts may be required.

b. All engines for the propulsion of land vehicles, except such tractor engines as are fitted with speed governors and operate normally under governor control, shall be tested with throttle wide open, with brake torque regularly varied from nothing to a maximum and back to nothing, at whatever speeds may result, and the horsepower-speed curve typical of the engine shall be determined instead of the horsepower at any given speed. In no case should a *safe* speed be exceeded and, when the highest speed used is the safe limit, it should be so stated. The maximum torque applied shall in all cases be greater than that which will produce maximum horsepower unless the speed required for this is too high to be safe.

Starting and Stopping.—Before recording the test observations, it is essential that an internal-combustion engine, together with all attachments and appurtenances, be brought to a condition of *steady state*. The test cannot be considered as having started until the engine has been in operation for a sufficiently long time to have attained its steady state for the conditions of the first run, and until preliminary observations have been made and recorded to prove that such steady state has been reached. If successive runs are to be made, each under some different conditions, these must be repeated for each run. In no case shall an engine be regarded as having attained its steady state unless jacket and lubricating-oil temperatures are substantially constant, nor shall steady state for the first run be regarded as established in a shorter time than 1 hour after the operating conditions have been imposed and before test observations have started. For successive runs at other conditions the minimum time to reach steady conditions for any engine shall be 10 minutes, the actual time being determined by the changes in conditions in successive runs. Maximum time to reach the steady state for any internal-combustion engine shall not be taken as greater than 24 hours, which may be regarded as permissible for the largest sizes of low-speed Diesel engines for motor ships and double-acting stationary gas engines.

Duration.—The duration of the test after the establishment of steady state for the set operating conditions shall be a matter of previous agreement but should be greater for engines where reliability is a question of importance and which require the longest time to reach the steady state. In no case shall the length of run be less than the period required to reach the steady state, subject to the additional condition that the length of run shall be great enough to insure the **accuracy of fuel measure-**

ment within 1 per cent, except when gaseous fuel is metered and when metering accuracy is not improved by lengthened runs.

General Data.—Instruments should be read at least quarter hourly when the conditions are uniform, and oftener when there is much variation. If there are wide fluctuations in readings, they should be shown by recording instruments. Indicator diagrams, if taken, must be taken from every working cylinder of multicylinder engines. Each indicator diagram should be marked when taken, with the number, date, time, corrected scale of spring, number and kind of cylinder, and end of cylinder, if double acting.

Fuel Consumption.—For liquid-fuel engines the actual fuel measured is stated in pounds in the report of the test, and is subject to no corrections.

For gaseous-fuel engines the cubic feet of gas at whatever pressure and temperature prevailed, indicated or recorded by the meter or calculated from the readings of flow-rate devices, are stated in the test report. This is to be corrected for the pressure and temperature at which the gas calorimeter burned the gas. The correction is to be made by multiplying the cubic feet of gas measured by the ratio of the absolute pressure at the meter to that at the calorimeter and by the ratio of the absolute temperature of the gas at the calorimeter to that at the gas meter. The product is the cubic feet of gas supplied to the engine measured at gas-calorimeter pressure and temperature.

Heat Consumption.—The number of heat units consumed by an engine per hour is found by multiplying the consumption by the heating value. The heating value for gaseous fuels is stated in the report of the test as found by the gas calorimeter, in B.t.u. per cubic foot, **high value**. The heating value of liquid fuels is stated in the report of the physical or chemical laboratory engaged for the determination, in B.t.u. per pound of fuel, **high value**.

Indicated Horsepower.—When the indicator can be used, and when in addition after using it the indicator diagrams are acceptable as to accuracy for determining mean indicated pressure, then the indicated horsepower shall be calculated as follows:

For **four-cycle engines** the indicated horsepower for **each single-acting** cylinder and for **each end** of each double-acting cylinder shall be found by using the formula

$$\text{i.hp.} = \frac{plan}{2 \times 33,000},$$

where p represents the mean indicated pressure, in pounds per square inch, l the length of the stroke, in feet, a the area in square inches of the piston less the area of the piston rod or tail rod, if any, and n the number of revolutions per minute. The total indicated horsepower of a double-acting cylinder is the sum of the horsepower developed in the

two ends, and the indicated horsepower of a multicylinder engine is the sum of the horsepower developed in all of the cylinders.¹

For **two-cycle engines** the indicated horsepower for each single-acting working cylinder, and for each end of each double-acting working cylinder, shall be found by using the formula

$$\text{i.hp.} = \frac{plan}{33,000},$$

where the symbols have the same meaning as above. The total indicated horsepower of all working cylinders is the sum of the horsepower developed in each of the cylinders, and in each double-acting cylinder it is the sum of the horsepower developed in each end.

The mean indicated pressure should be found by dividing the area of the diagram in square inches, as determined with a correct planimeter, by the length of the diagram, in inches, and multiplying the quotient by the average corrected scale of the indicator spring.

Accuracy tests must be applied to determine the acceptability of the mean indicated pressure thus determined, especially when cylinder pressures and spring scales are high (as for injection oil engines) and card areas small, as is usual for two-cycle engines of small size.

Brake Horsepower.—The brake horsepower is found by multiplying the net force w , in pounds, on the brake arm by the circumference of the circle whose radius is the horizontal distance l , in feet, between the center of the shaft and the bearing point or weight center at the end of the brake arm, and by n , the number of revolutions of the brake shaft per minute, and dividing the final product by 33,000.

$$\text{b.hp.} = \frac{2 \times 3.1416lwn}{33,000}.$$

Brake Mean Effective Pressure.—Brake mean effective pressure is calculated for A.S.M.E. Codes from the brake horsepower. The formula for four-cycle engines is

$$\text{b.m.e.p.} = \frac{\text{b.hp.} \times 33,000 \times 2}{lan}.$$

The formula for two-cycle engines is

$$\text{b.m.e.p.} = \frac{\text{b.hp.} \times 33,000}{lan}.$$

Thermal Efficiency.—The fraction of the heat consumption converted into work is the “thermal efficiency” and is found by dividing 2,545 (B.t.u. equivalent to 1 horsepower-hour) by the number of heat units

¹ Gas or oil engines operating by “hit-and-miss” governing do not have an explosion in every cycle and the number of explosions must be used for the value of $n/2$, in the foregoing formula.

actually supplied per brake or per indicated horsepower-hour. The quotient is multiplied by 100 to express the thermal efficiency in percentage. The formulas are

$$\text{Brake thermal efficiency, } E_b = \frac{2,545}{Q_b},$$

$$\text{Indicated thermal efficiency, } E_i = \frac{2,545}{Q_i},$$

where

Q_b = B.t.u. per brake horsepower-hour.

Q_i = B.t.u. per indicated horsepower-hour.

The two terms Q_i or Q_b are equal, respectively, to the **high heating value** of the fuel (B.t.u. per pound or cubic foot) multiplied by the amount of fuel supplied per indicated horsepower-hour (pounds or cubic feet) or the same quantity multiplied by the amount of fuel supplied per brake horsepower-hour.

Data and results should be reported in accordance with the form of Table XXII for horsepower tests made at constant speed, and of Table XXIII for horsepower tests made over the whole speed range of a variable-speed engine.

TABLE XXII.—DATA AND RESULTS OF INTERNAL-COMBUSTION ENGINE TEST AT CONSTANT SPEED

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

Description, Dimensions, Etc.

7. Type of engine, two or four cycle, single or double acting, horizontal or vertical; if four cycle, the valve arrangement, L, T, or I head¹; if two cycle, the type of scavenging; if single acting, whether cross-head or trunk piston; if multicylinder, the arrangement of cylinders and cranks, gas or liquid fuel; and if liquid fuel, carbureter type or other class name fixing manner of treating fuel, such as air injection, mechanical injection, or primary combustion.....
8. Class of service, stationary and special feature, marine, aircraft or vehicle, and what kind.....
9. Auxiliaries attached, such as magneto, fuel-injection pump, fuel-circulating pump, lubricating-oil pumps, jacket circulating pumps, scavenging pumps, spray-air compressor, maneuvering or starting air compressor, radiator fans, oil or fuel coolers or heaters.....
10. Auxiliaries, independent or separately driven, and power.....
11. Details of type, capacity, maker's name, and other features of auxiliaries.....
12. Rated brake horsepower of engine, or kilowatt of electric generating set, and speed
13. Grade of fuel for which engine is designed, kind of gas or specification of liquid fuel, and what was used in test.....
14. Special structural features for fuel utilization.....

¹ See Moyer, "Gasoline Automobiles," 3d ed., p. 51, McGraw-Hill Book Company, Inc., New York.

15. Special structural features of speed and power control, and governor or reversing gear.....
- Number of working cylinders.....
16. Diameter of all cylinders working, four or two cycle.....in.
17. Bore and stroke of power cylinders.....in. or ft.
18. Diameter of piston and tail rods of power cylinders.....in.
19. Head-end horsepower constant for power cylinders (stroke $\times$ net piston area $\div$ 33,000).....
20. Crank-end horsepower constant for power cylinders (stroke $\times$ net piston area $\div$ 33,000).....
21. Capacity of generator or other apparatus consuming power of engine.....
22. Characteristics of generator—direct current or alternating current, volts, cycles, phase.....

Test Data and Results

23. Duration of test.....hr.

Pressures, Average

24. Barometric pressure.....in. of mercury.....lb. per sq. in.
25. Spray-air pressure (air injection Diesel engines) average (say whether gage or absolute).....lb. per sq. in.
26. Exhaust back pressure at exhaust nozzle.....in. of water
27. Jacket-water supply pressure at crank-shaft center.....lb. per sq. in.
28. Manifold vacuum (carburetor engines).....in. of water
29. Gas pressure at meter (gaseous fuel).....in. water.....lb. per sq. in.
30. Lubricating-oil pressure, circulating forced-feed system at crank-shaft center
.....lb. per sq. in.
31. Scavenging-air pressure (gage), average in engine header, two-cycle
engines.....lb. per sq. in.

Temperatures

32. Engine-room temperature.....°F.
33. Temperature of fuel at meter.....°F.
34. Temperature of main air supply entering engine.....°F.
 Location of thermometer.....
35. Temperature of mixture at intake port (carburetor engine).....°F.
36. Temperature of main jacket-water or oil inlet.....°F.
37. Temperature of main jacket-water or oil outlet.....°F.
38. Temperature of piston cooling water or oil at inlet.....°F.
39. Temperature of piston cooling water or oil at outlet.....°F.

Fuel Properties

40. Heating value of gas, high value at standard temperature and
 pressure (p. 241).....B.t.u. per cu. ft.
41. Heating value of liquid fuel, high value.....B.t.u. per lb.

Total Quantities

42. Total gaseous fuel at meter pressure and temperature.....cu. ft.
43. Correction factor for gas = (absolute pressure at meter $\times$ absolute standard
 temperature) $\div$ (absolute standard pressure $\times$ absolute temperature at meter)
44. Total gaseous fuel at standard pressure and temperature (Item 42 $\times$ Item
 43).....cu. ft.
45. Total liquid fuel.....lb.

Hourly Quantities

46. Total gaseous fuel per hour at standard pressure and temperature.....cu. ft.
 47. Total liquid fuel per hour.....lb.

Heat Consumption

48. Total heat in fuel supplied per hour (Item 46 $\times$ Item 40, or Item 47 $\times$ Item 41).....B.t.u.

Indicator Diagrams

49. Mean indicated pressure, average of all power cylinders.....lb. per sq. in.
 50. Maximum pressure of all power cylinders.....lb. per sq. in.

Speed

51. Revolutions per minute.....r.p.m.
 52. Piston speed (mean of power pistons).....ft. per min.

Power

53. Indicated horsepower of all working cylinders.....i.hp.
 54. Brake horsepower developed by whole engine by brake or dynamometer measurement.....b.hp.
 55. Brake mean effective pressure (p. 417).....lb. per sq. in.
 56. Horsepower input to motor or output of engine driving independent scavenging pump.....hp.
 57. Horsepower input to motor or output of engine driving independent injection air compressor.....hp.
 58. Horsepower input to motors of other independent auxiliaries essential to engine operation.....hp.
 59. Net or actual brake horsepower of engine (Item 54 — Items 56, 57, and 58) . . b.hp.
 (a) Net brake mean effective pressure.....lb. per sq. in.
 (b) Net torque at 1-ft. radius equivalent to brake horsepower.....ft.-lb.

Economy Results

60. Fuel consumption per indicated horsepower-hour, gas (Item 46 $\div$ Item 53), or liquid (Item 47 $\div$ Item 53).....cu. ft. or lb.
 61. Fuel consumption per brake horsepower-hour,¹.....cu. ft. or lb.
 62. Heat consumed per indicated horsepower-hour (high heat value—p. 241) . . B.t.u.
 63. Heat consumed per brakepower horsepower-hour¹ (high heat value).....B.t.u.

Efficiency Results

64. Indicated thermal efficiency (2,545 $\div$ Item 62).....per cent
 65. Brake thermal efficiency (2,545 $\div$ Item 63).....per cent

Specimen Diagrams

66. Sample diagram from each power cylinder.....

Electrical Data

67. Average volts each phase.....volts
 68. Average amperes each phase.....amp.
 69. Total electrical output corrected for winding 1, 2, or 3 phase.....kva.
 70. Power factor.....per cent
 71. Total output, in kilowatts.....kw.
 72. Separate excitation.....kw.

¹ According to conditions the brake horsepower will be either Item 54 or Item 59.

73. Net electric output, in kilowatts.....kw.
 (a) Field voltage.....volts
 (b) Field current.....amp.

Electric Economy and Efficiency Results

74. Fuel consumed per net kilowatt-hour (Item 46 ÷ Item 73) or (Item 47 ÷ Item 73).....cu. ft. or lb.
 75. Heat consumed per net kilowatt-hour (Item 48 ÷ Item 73).....B.t.u.

TABLE XXIII.—DATA AND RESULTS OF INTERNAL-COMBUSTION ENGINE TEST AT VARIABLE TORQUE AND SPEED

General Information—Same as for Table 1
Description, Dimensions, Etc.

7. Type of engine, carbureter or oil, single or double acting; if four cycle, the valve arrangement, L, T, or I head; if two cycle, the type of scavenging; if single acting, whether cross-head or trunk piston; if multicylinder, the cylinder and crank arrangement; and if radial cylinders, whether cylinders are fixed or rotate; whether gas or liquid fuel; and means of fuel utilization by class name, such as Diesel or solid injection, for example.....
 8. Class of service, aircraft, automobile, truck, tractor, railroad or marine, for gear, electric, or direct propeller drive, single direction or reversing.....
 9. Auxiliaries, attached and independent, kind, maker's name, capacity, etc.....
 10. Rated horsepower and speed, if any, or speed at maximum torque...hp...r.p.m.
 11. Maximum safe speed.....r.p.m.
 12. Bore and stroke of power cylinders.....in. or ft.
 13. Diameter of pistons and tail rods of power cylinders.....in.
- | | Test No. | | |
|---|----------|------|------|
| | 1 | 2 | 3 |
| 14. Brake horsepower.....b.hp. | | | |
| 15. Speed.....r.p.m. | | | |
| 16. Total fuel.....lb. | | | |
| 17. Duration of run.....hr. | | | |
| 18. Fuel per hour.....lb. | | | |
| 19. Fuel per hour per brake horsepower.....lb. | | | |
| 20. Heating value of fuel (high value).....B.t.u. per lb. | | | |
| 21. Heat consumed per hour (Item 18 × Item 20).....B.t.u. | | | |
| 22. Heat consumed per hour, per brake horsepower (Item 21 ÷ Item 14).....B.t.u. | | | |
| 23. Brake thermal efficiency (2,545 ÷ Item 22).....per cent | | | |
| 24. Brake mean effective pressure (p. 417).....lb. per sq. in. | | | |
| 25. Torque at 1-foot radius equivalent to brake horsepower ft.-lb. | | | |
| 26. Curve of revolutions per minute plotted horizontal—against vertical, as follows:..... | | | |
| (a) Brake horsepower (Item 14).....b.hp. | | | |
| (b) Fuel per hour (Item 18).....lb. | | | |
| (c) Fuel per hour per brake horsepower (Item 19).....lb. | | | |
| (d) Brake thermal efficiency (Item 23).....per cent | | | |
| (e) Brake mean effective pressure (Item 24).....lb. per sq. in. | | | |
| (f) Torque at 1-ft. radius (Item 25).....ft.-lb. | | | |

A.S.M.E. TEST CODE (1928) FOR GAS PRODUCERS (ABRIDGED)

This Code for conducting tests of gas producers is intended primarily for tests of producers whose gas is to be used for power purposes but

also may be used for those generating gas for metallurgical and other heating purposes. The term "fuel," as herein used, includes all possible fuels—coals, coke, charcoal, refuse, and oils. In testing a gas producer, tests of the auxiliary apparatus must be frequently included, as being essential parts of the gas-generating unit. If a complete test of the gas-producer plant is desired, separate records should be made of the amounts of fuel, water, manual and skilled labor, electric current, and of the power required to operate the producer and each of its auxiliaries, the cost of each of same, and the usual calculations should be made and the proper conclusions drawn.

Operating Conditions.—The method of testing, the length of the test, and the operating conditions necessary to conform to the object of the test should then be determined. Care should be taken to see that these conditions are maintained throughout the test. With a suction producer, tests should be made for air leaks which should be stopped.

The character of the fuel to be used should be determined. If an untried fuel is selected, a preliminary trial of the fuel in this apparatus should be made and the best method of using it ascertained. Whatever the fuel, it should be weighed and not measured. It should be sampled for dryness and due allowance made for the moisture used in producing gas during the test.

In tests of maximum efficiency and capacity of a producer for comparison with other producers, the fuel should be of the kind which is commercially regarded as a standard for such use in the locality where the test is made. Unless previously decided otherwise, the fuel selected for test of the gas producer should be the best of the class so selected, free from unusual slag-forming impurities, and of the proper and uniform size for its specific use.

Duration of Test. Full-time and Complete Tests.—The duration of both efficiency and capacity full-time tests of a producer, with the exceptions noted below, should be such that the probable error in weighing the coal does not exceed 2 per cent, or that the total consumption of fuel is at least twice the weight of the fuel contained in the producer when in normal operation, this weight being estimated in the case of coal at 45 pounds per cubic foot. In no case should the duration be less than 24 hours after the producer has reached the desired and constant operative temperatures and conditions, except by previous agreement. Tests to determine the clinkering tendencies of a fuel and the quality of the ashes will normally require five or more days in large producers.

Intermittent Producers.—In cases which require the fuel bed to be entirely removed and rebuilt at regular intervals, and in producers where a complete cleaning and renewal occur before the total consumption above stipulated has been reached, the duration should be that of one or more of the regular commercial operating cycles, or the time

elapsing between a corresponding number of successive renewals of the fuel bed. The time of starting and stopping may be taken from either immediately after cleaning, or midway between two cleanings, care being taken in the latter case to see that the fuel bed is at the same level in stopping as in starting.

Spot Tests.—Where the cost of making a full-time test is prohibitive, or where for any other reason it is not desired, or where a frequent check is desired on the capacity, efficiency, and character of the output of the gas producer at any time, a “spot test” may be made without elaborate preparation, with the use of equipments frequently found in such plants, and with the services of an analytical chemist, provided (a) the fuel used, such as anthracite, charcoal, coke, and oil, produces negligible amounts by weight of tar or other condensible carbon vapors; and (b) the gas cooling water is recirculated so that it remains saturated with the carbon dioxide and other gases being produced.

For this purpose there will be needed an appliance for continuously sampling the gas, say, for 1 hour or else for taking a series of spot samples; apparatus for analyzing and determining the composition of the gas produced; and either a continuous gas calorimeter or the equivalent. The average heating value of the gas may be calculated from the volumetric analysis of the gas under standard conditions by the use of appropriate values of the calorific power of the constituent gases. The average volumetric analysis for the included period is then converted into an analysis by weight, from which the total weight of carbon in 1 cubic foot of standard gas may be determined as follows: If the sum of the proportions by volume of carbon monoxide, carbon dioxide, methane, and other carbon compounds containing one atom of carbon be multiplied by 0.03127, and if the sum of the proportions by volume of the C_2H_4 , C_2H_6 , and other carbon compounds containing two atoms of carbon be multiplied by 0.6298 (nearly twice 0.03127), and the two products added together, the sum will be the weight of carbon in 1 cubic foot of the gas at 68°F. and 29.92 (approximately 30 inches) of mercury at 32°F.¹ = C_s . The ultimate analysis and the calorific value of the coal as actually fired to the producer and an analysis for the carbon content of the dried ash will be necessary.

Starting and Stopping.—The conditions regarding the temperature of the producer and its contents, and the quantity and quality of the latter, especially of the clinker, should be as nearly as possible the same throughout the test, and particularly so at the beginning and end of the trial. To secure the desired equality of conditions, the starting and stopping should occur at the conclusion of the times of regular cleanings, and they should be preceded for a period of not less than 8 hours by the same regular working conditions as are intended to characterize the test as a whole. Unless the condition of the fuel bed at the beginning and end of a test can be accurately determined, and possible differences in level allowed for, so that the error in determining the net weight of fuel used during the test shall not exceed 2 per cent, the tests should be abandoned as valueless. The operations of starting and stopping should be carried on as follows:

¹ The standard pressure of 29.92 inches of mercury at 32°F. is nearly the same as 30.03 inches of mercury at 68°F.

Continuous Producers with Grate and No Ash Bed.—Remove the ash and clinkers from the grate and the lower part of the furnace space, taking care that the crust or closely united layer which supports the fuel above is not unduly disturbed. Then break open the crust and allow the mass to drop into the space left vacant. Introduce a poker rod through the poker holes in the upper head and stir up the fuel within, thereby causing it to settle and fill up the remaining spaces. As a final step, quickly replenish the producer with fuel to the working depth, which should be measured, fill the hopper level full, take the time, and consider this the starting time. Then clean the ashpit, and thereafter proceed with the regular work of the test, using weighed fuel.

When the time arrives for bringing the trial to a close, the cleaning operations described above should be repeated, ending with filling the hopper, taking the time, and considering this the stopping time; finally, the ashes and refuse should be removed from the ashpit, weighing and sampling being done as directed, and the producer replenished with fuel to the same working depth as at the start.

Continuous Producers with Supporting Ash Beds.—Remove the ashes until the top of the ash bed is lowered to the normal working point. Introduce the poker rod and break down any bridge or crust that may have formed, at the same time closing up the channels that run through the fuel bed, thereby making the bed homogeneous. Then replenish the producer with fuel to the working depth, which should be measured, to the same working depth as at the start, fill the hopper level full, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test, using weighed fuel.

When the time approaches for ending the test, the operations above described are repeated, ending with replenishing the producer and filling the hopper with weighed fuel, taking the time, and considering this the stopping time. The ashes and refuse finally removed, unless already dry, are to be dried before weighing, if they have been wetted after being removed; otherwise they should be weighed both wet and dry and the water accredited to the water used during the test, or a sample should be taken and the moisture, as determined therefrom, allowed for.

Intermittent Producers.—Thoroughly clean the producer of its entire contents. Introduce a weighed supply of fuel, start the fire, and build up the fuel bed to its working condition, using weighed fuel. When this point is reached, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test.

When the time approaches for ending the test, burn the fuel bed as low as practicable to prepare for cleaning, note the time, and consider this the stopping time. Then completely empty the producer, quench the fire remaining in the live coals, separate and weigh the cake and ash, and deduct the weight of the former from that of the fuel as charged. Finally, dry the ash and refuse or take a sample and allow for the moisture as determined therefrom.

Some idea of the thickness of the ash-bed, incandescent-fuel, and green-fuel zones may be obtained by the simultaneous use of several rods made of $\frac{3}{4}$ -inch pipes, having a stop pin 3 inches long inserted 2 feet from the upper and handled end, and long enough to extend down into the ashpit. These rods should be inserted vertically, through observation holes, or poke holes, in the top of the producer. The top level of the green-coal zone and the height of the fuel in the producer at each place will be shown by the distance that the rod is inserted below the observation hole when resting on the fuel. When forced down through the fuel and to the stop pin and left there for 2 or 3 minutes, and until part of its length gets red hot, when withdrawn, the red-hot part will show the thickness and position of the incandescent or fire zone. The depth of the ash zone will be the distance from the lower line of visible heat in the rod to the grate blower hood, or bottom of the producer. The green-fuel zone will be the space above the incandescent zone. Its thickness will be the difference between the distance from

the top of the observation hole to the top of the fuel and to the incandescent zone. With tarry fuel, it may be indicated by the tarry deposit on the rod. The gas zone may be noted by the sooty deposit on the upper end of the rod. The average results should be the ones reported.

The depth of the fuel bed is measured by the distance from the lower line of visible heat in the rod to the top of the green fuel in the producer.

The distance from the top of the producer to the surface of the fuel bed may be found by direct measurement with the poker rod, by noting by sense of touch when the end of the rod reaches the fuel.

The product of the proportion of ash from the sample of coal as fired and the proportion of carbon to net ashes from the sample of ashes and refuse will be the proportion of coal in the carbon of the ashes and refuse and will be the weight of the carbon lost in the ashes per pound of coal as fired. The carbon lost in the ashes taken from the carbon in the coal as fired will give the weight of carbon passing over in the gas per pound of coal as fired.

If C_1 is the carbon in 1 pound of coal as fired, C_2 is the carbon of 1 pound of the ashes and refuse from the producer, and A_1 is the ash in 1 pound of the coal, by analysis, then $\frac{C_2}{1 - C_2}$ is the carbon in 1 pound of the net ash, and $A_1 \times \frac{C_2}{1 - C_2} = C_3$ will be the carbon from 1 pound of coal lost in the ashes, and $C_1 - C_3 = C_4$, the carbon gasified per 1 pound of coal. The cubic feet of gas made per pound of coal will be the quotient obtained by dividing the carbon gasified per pound of coal by the weight of carbon in 1 cubic foot of the gas, or $C_4 \div C_5$.

The cubic feet of gas per pound of coal multiplied by the B.t.u. per cubic foot of the gas will give the B.t.u. of output per pound of coal. The *cold gas* efficiency of gas making will be

$$E = \frac{C_4 \times \text{B.t.u. per cubic foot of gas}}{C_5 \times \text{B.t.u. per pound of coal}}$$

The desired output in B.t.u. per hour divided by the product of the efficiency and the B.t.u. per pound of coal will give the pounds of coal which will have to be fired per hour to generate the desired output. Weighing the coal input over any desired period at steady load will therefore permit the determination of the output capacity.

Spot Tests with Tar-producing Fuels.—Spot tests may be made when bituminous coals are used, if the tar and soot per cubic foot of the outlet gas are determined. With plants operating on tar-producing and soot-producing fuels, the foregoing methods will apply only where the cooling water is recirculated and the tar and soot are either gasified or separated and determined per foot of gas delivered. Where the tar is wasted, it must first be collected, weighed, sampled, and its weight compared with the weight of fuel used during the same time, so as to determine the amount of tar lost per pound of coal gasified. If the tar and soot per cubic foot of gas are carefully determined, the spot-test method can be used with the uncleaned and hot gases from pressure producers.

In case the water is not recirculated, there will be a considerable absorption of carbon dioxide by the cold water used in washing the gas. The carbon so lost can be determined either by comparing the carbon dioxide contents of the gas samples taken simultaneously before and after passing the coolers, or by metering the cooling water used and determining by analysis the amount of carbon carried away as carbon dioxide in the water.

With the use of continuously recording gages, calorimeters, and weighing and sampling devices, it is possible with producers using anthracite coal or charcoal to make spot tests and determine the instantaneous capacity and economy.

Because of the simplicity of the equipment required, the decrease in the number of observations to be made, the decrease in the time consumed, and the ability to get quick results with fair accuracy and with no greater precaution than in the conventional heat-balance method, this method of testing a gas producer is commended.

Short Tests.—If less complete and less expensive tests are desired than the full time tests in Table XXIV (page 430), they may be made by determining the following items: the weight of the dry fuel gasified (115), the ultimate analysis of the fuel (85 to (90), the proximate analysis (80) to (84), the calorific values of the fuel and gas (107) to (111), the temperature and humidity of the air (64) and (51), the pressure, humidity, and temperature of the steam used (42), (52), and (79), the analysis of the refuse (91) to (95), the temperature of the gas leaving the producer (65), the analysis of the gas made including the amounts of tar and moisture (96) to (106), (129), (138) and (130).

It will not be necessary in these tests to meter the steam used or the gas made. The volume of gas made is determined by knowing the amount of carbon gasified per hour and the average amount of carbon in 1 cubic foot of gas as it leaves the producer, including the carbon in the tar and soot. The amount of water vapor coming from all sources must equal the water equivalent to the hydrogen, both free and combined, in the gas, tar, and water going into the gas main from the producer. If the amount of hydrogen coming from the fuel and the amount coming from the moisture in the air are known, the remainder will equal the hydrogen entering the producer as steam from the boiler, and as vapor from the ashpit, water seal, and elsewhere. From these data the gas and tar produced per ton of coal and the total heat in the gas, both sensible and potential, and the thermal efficiency can be calculated.

Analyses and Calorific Tests of Fuels.—The quality of the gas should be determined by calorimetric test or chemical analysis or both. Continuous samples for the purpose should be taken from the delivery pipe at a point near the producer and at other points as may be needed. The calorimetric test should be made with a Junkers type of calorimeter (page 239). For an approximate determination of the composition of the clean gas, the Orsat apparatus (page 272) may be used, and for a complete determination the Hempel apparatus (page 276) or its equivalent. The frequency with which these determinations should be made depends on the uniformity of the output.

Measuring the Steam.—The steam used may be measured by weighing the water before it is evaporated, provided there are no leaks or other uses made of the steam so generated, or by a suitable steam meter, or by the use of the Napier formula (page 195).

Total Volume of Gas Delivered.—The volume of gas delivered should be calculated from the chemical analyses of the fuel and gas. It may be measured by the Pitot meter¹ (page 184) or by the orificemeter (page 190) and will be determined by multiplying the effective area of the delivery pipe in square feet at the tube by the average velocity of the gas in feet per minute over that area, and the product by the duration of the trial in minutes.

The equivalent volume of the gas at **standard** atmospheric pressure of 29.92 inches of mercury (at 32°F.) and the standard temperature of 68°F. is obtained by multiplying the measured volume by the absolute pressure of the gas in pounds per square inch (gauge pressure plus the equivalent barometric pressure) and by the con-

¹ Care must be taken to keep the Pitot tubes and orifice free from accumulations of soot and tarry matter by intermittently blowing or cleaning the tubes, preferably with steam hot enough to melt off the tar, or by the use of jets of steam at high temperature placed one or more diameters ahead of the orifice and impinging on the orifice diaphragm.

stant 35.89 $\left(= \frac{527.6}{14.70} \right)$, and dividing the product by the absolute temperature of the gas [temperature by thermometer plus 460°F. (more accurately 459.6°F.)]. Barometer corrections are given on page 4.

The gas as it leaves the scrubber is saturated with water vapor at the pressure due to its temperature, and this pressure is to be deducted from the absolute pressure, as observed, to obtain the net pressure of dry gas. The volume of dry gas in that case is obtained by the following formula:

$$V = V_o \times \frac{P - P_w}{P_a} \times \frac{527.6}{t + 460},$$

in which V = equivalent volume of dry gas at 68°F. and 29.92 inches of mercury at 32°F. leaving scrubber.

V_o = observed volume of gas leaving the scrubber.

P = absolute pressure at the point where the volume is measured.

P_w = pressure of saturated water vapor at temperature t .

P_a = standard atmospheric pressure (29.92 inches of mercury or approximately 30 inches) at 32°F., equivalent to 14.70 pounds per square inch.

t = observed temperature of gas leaving scrubber degrees Fahrenheit.

P and P_w are to be taken in the same units as P_a .

The weight of dry gas delivered by a gas producer can be obtained by calculation from the analyses of the fuel and of the dry gas and from the net weight of the carbon gasified, by the following method.

The weight of dry gas per pound of net carbon gasified W (1 pound less than the sum of the weights of the carbon in the tar and soot delivered in the raw gas, and the carbon in the ashes per pound of carbon in the fuel) is given by the equation below.

$$W = \frac{11\text{CO}_2 + 8\text{O}_2 + 7(\text{CO} + \text{N}_2 + \frac{1}{2}\text{H}_4) + 4\text{CH}_4 + 0.5\text{H}_2}{3(\text{CO}_2 + \text{CO} + \text{CH}_4 + 2\text{C}_2\text{H}_4)}$$

in which carbon dioxide, oxygen, carbon monoxide, hydrogen, methane, C_2H_4 , and nitrogen are percentages by volume of the dry gas per pound of net carbon gasified. Multiplying the pound of dry gas per pound of net carbon gasified by the percentage of net carbon in the fuel (= percentage of carbon in the fuel less the sum of the percentages of carbon in the tar and soot delivered in the raw gas and the carbon in the ashes per pound of fuel) and dividing by 100 gives the weight of dry gas per pound of fuel burned. The volume of gas per pound of fuel is found by dividing this weight by the weight of 1 cubic foot of the gas calculated from Table XXIII A (page 428).

The weight of carbon per 100 cubic feet of gas at 68°F. and 29.92 inches of mercury at 32°F. can be found directly from the analysis by multiplying the percentage of volume of each gas containing carbon by the pounds of carbon in 1 cubic foot of the gas as follows:

$$\begin{aligned} (\text{CO} + \text{CO}_2 + \text{CH}_4) &\times 0.03127 \\ \text{C}_2\text{H}_4 &\times 0.06284 \\ \text{C}_2\text{H}_6 &\times 0.06313 \end{aligned}$$

and adding the results together. The volume of clean gas delivered per pound of fuel may be found by subtracting the sum of the weights of carbon in the tar, soot and ashes, and refuse per pound of fuel from the weight of carbon in a pound of fuel, and dividing the remainder by the weight of carbon in a cubic foot of clean, dry gas, all at the same temperature and pressure.

Net Volume of Dry Gas Delivered.—The net volume of dry gas delivered is found by subtracting from the total volume the volume of dry gas that would be required for

furnishing steam or power for any purposes concerned in the operation of the producer and its auxiliaries.

Gross Weight of Dry Gas.—The weight of dry gas delivered is found by multiplying the volume in cubic feet reduced to 68°F. and 29.92 inches of mercury by the weight per cubic foot of dry gas, which is found by multiplying the percentage of each component gas as found by analysis by its weight in pounds per cubic foot at 68°F. and at 29.92 inches of mercury at 32°F. as given in the following table (calculated from Landolt and Bornstein's figures at 32°F. and 29.92 inches) and dividing the sum of the products by 100.

TABLE XXIII A.—WEIGHT OF GASES
Pounds per Cubic Foot at Standard Conditions

H ₂	0.00522	CH ₄	0.04169
O ₂	0.08312	C ₂ H ₄	0.07331
N ₂	0.07274	C ₂ H ₆	0.07891
CO.....	0.07273	SO ₂	0.17025
CO ₂	0.11499	H ₂ S.....	0.08846

Calorific Value of Gas.—The calorific value of the dry gas per cubic foot is obtained by means of a Junkers type of calorimeter (page 239), but it may also be obtained from the analysis by multiplying the percentage of each combustible constituent gas by its heating value per cubic foot at 68°F. and 29.92 inches of mercury as given below:

CALORIFIC VALUE PER CUBIC FOOT OF DRY GAS

CO.....	318.8	CH ₄	1,002
H ₂	320.2	C ₂ H ₄	1,595

These figures are the product of the foregoing values of weight of gas per cubic foot by the heating value of 1 pound of gas, according to Thomson.

Moisture in Gas.—The moisture in the gas leaving the producer is found by passing a measured sample of the gas from which all tarry vapors have been removed through a heater so as to keep the temperature up to what the temperature of the gas was when it left the producer or not less than about 250°F. and then through a chloride of calcium tube and weighing the amount of moisture absorbed.

The moisture in the gas leaving the scrubber is best found by calculation, assuming that the gas is saturated with moisture.

Percentage of Tar and Soot in Gas.—The percentage of tar and soot is found by comparing the total weight determined, including that collected from the various tar drips, with the total weight of dry fuel used.

Sensible Heats.—The sensible heat of the gas as delivered is the sum of the products of the constituents of the gas delivered per pound of dry fuel by their respective volumetric specific heats and the total heat of the moisture content above 68°F. The sensible heat of coal, for example, is the product of its specific heat per pound (0.20 to 0.24 B.t.u.) by the temperature difference of the coal, or surrounding air above or below 68°F.

Efficiencies.—The efficiency is the ratio of the total heat carried by the gas (both the heat of combustion and the sensible heat above 68°F.) to the sum of the heats supplied in its production. This is the "hot-gas efficiency" (page 429). In the case of hot and uncleaned gases, the total heat carried by the gas will include the heats of combustion of the constituent gases and of the carbonaceous and tarry vapors, liquids, and solids, their superheats above the temperature corresponding to their partial pressures, the latent heats of the constituent vapors including the water vapor, and their sensible heats above 68°F.

When the producer is supplied with steam from an outside source, rather than from its own vaporizer, for the generation of gas, the total heat used to generate the steam so supplied should be added to the heat of the fuel supplied in the same length of time to determine the efficiency.

The combustible is determined by subtracting from the weight of charged fuel the moisture in the coal and the weight of ash refuse and unburned fuel withdrawn from the producer or ashpit during or at the end of the test. The "combustible" used for determining the calorific value is the weight of the fuel less the moisture and ash found by analysis.

The "efficiency of conversion and cleaning" or "gross efficiency" in the foregoing calculation is found by using the total gross volume of dry gas delivered. The "efficiency of the plant" or "net efficiency" is found by using the net volume of dry gas delivered. By "net volume" is meant the gross volume less the actual gas used, or the equivalent, to operate the auxiliaries; that is, the net volume is the amount of gas available for sale or for uses outside of the producer plant.

The *cold-gas efficiency* is the sum of the products of the constituent parts of the cold gas delivered per pound of dry fuel by their higher heating values divided by the sum of the heating value of a pound of the fuel used in the producer and the heating value of the fuel used to generate the steam supplied from an outside source per pound of the fuel used in the producer.

The *hot-gas efficiency* is obtained by adding to the potential heat of the cold coal gas as above stated the sensible heat of the gas as determined (page 428), and dividing by the sum of the higher heating value of a pound of the fuel and the heating value of the fuel used to generate the steam supplied from an outside source per pound of the fuel used in the producer.

Unless otherwise stated, the higher, or gross, heating (calorific) value should always be used. If the lower, or net, heating value is used, it should always be noted, so as not to cause confusion in results.

The "cold-gas" and "hot-gas" efficiencies, both "gross" and "net," should be determined for both the higher and the lower heating (calorific) values for the gas.

Heat Balance.—The various quantities showing the distribution of heat in the heat balance given in the table (page 434) are computed in the following manner:

The heating value of the dry gas is found by multiplying the cubic feet of gas at 68°F. and 29.92 inches of mercury at 32°F. that is generated per pound of dry fuel by the higher heating value of 1 cubic foot of gas at 68°F. and 29.92 inches of mercury at 32°F.

The sensible heat in the dry gas is found by multiplying the weight of dry gas generated per pound of fuel by the mean specific heat of the gas and by its temperature measured above 68°F.

The heat carried away by the scrubber is obtained by multiplying the weight of water fed to the scrubber by the number of degrees rise of temperature and dividing the product by the total weight of dry fuel consumed in the same time.

The heat contained in the moisture leaving the producer is found by multiplying the total weight of dry gas per pound of dry fuel by the proportion of moisture in the gas and by the total heat of 1 pound of superheated steam at the temperature of the gas leaving the producer reckoned from 68°F. and at its partial pressure (page 247).

The loss due to combustible matter in the ashes and refuse is found by multiplying the proportion that this combustible bears to the whole amount of dry fuel by 14,544 B.t.u. The heating value of the combustible matter in the ashes and refuse should be determined by chemical analysis or by a calorimeter, as a check.

In the case of those producers which are supplied with steam from a separately fired boiler rather than solely from the vaporizer of the producer from the sensible heats of the gases generated, either the fuel used in operating the boiler per pound of

the fuel used in the producer or the total heat of the steam as delivered to producer, divided by a known, or assumed, efficiency of the boiler, should be included in the potential or available heat energy delivered to the producer, and should be used in calculating the efficiencies.

TABLE XXIV.—DATA AND RESULTS OF GAS-PRODUCER TEST

General Information

(1) Date of test; (2) location; (3) owner's name; (4) builder's name; (5) test conducted by; (6) object of tests.

Description, Dimensions, Etc., of Gas Producer

7. Type of producer.....	
8. Rated size, power, or capacity of producer.....	
9. Duration of test.....	hr.
10. Trade name, kind, and size of solid fuel.....	
11. Trade name and kind of oil fuel.....	
(a) Oil fuel, specific gravity at 68°F.....	° Baumé, A.P.I.*
(b) Oil fuel, specific gravity.....	referred to water
(c) Oil fuel, weight per gallon at 68°F.....	lb.
12. Outside diameter and height of producer.....	ft.
13. Inside diameter at hearth.....	ft.
14. Maximum inside diameter of lining of producer.....	ft.
15. Minimum inside diameter of lining of producer.....	ft.
16. Actual net diameter of the gasification zone.....	ft.
17. Height from lowest level of grate to crown.....	ft.
18. Height from lowest level of grate to center of gas outlet.....	ft.
19. Vertical distance between highest and lowest levels of inclined grate.....	ft.
20. Gross diameter of grate.....	ft.
21. Corresponding projected area of grate.....	sq. ft.
22. Gross diameter of central air inlet through grate.....	ft.
23. Corresponding area of same.....	sq. ft.
24. Gross area of grate, if inclined.....	sq. ft.
25. Area of air spaces in grate.....	sq. ft.
26. Ratio of air spaces in grate to net, or gross, grate area.....	
27. Diameter of steam-blast inlet.....	in.
28. Corresponding area of same.....	sq. in.
29. Type of blower used.....	
30. Diameter of steam pipe to blast inlet, or blower.....	in.
31. Diameter of gas outlet.....	in.
32. Corresponding area of same.....	sq. in.
33. Area of fuel bed at maximum diameter.....	sq. ft.
34. Actual net area of fuel bed at gasification zone.....	sq. ft.
35. Area of water heating surface in vaporizer or jacket.....	sq. ft.
36. Method of poking producer.....	
37. Method of rotating producer.....	
38. Method of generating steam for producer.....	
39. Method of cooling gas.....	
40. Method of scrubbing and purifying gas.....	
41. Method of supplying water for cooling and cleaning.....	

*Baumé scale, A.P.I., refers to the gravity scale adopted by the American Petroleum Institute (p. 522).

*Test Data and Results***Pressures**

42. Steam pressure in boiler, or vaporizer, or of steam as delivered to producer, by gage. lb. per sq. in.
43. Gas pressure, or suction, in main at point where gas is measured ...in. of water
44. Pressure, or suction, at top of producer.....in. of water
45. Pressure, or suction, beyond scrubber.....in. of water
46. Pressure, or suction, beyond purifier.....in. of water
47. Draft pressure, or suction, in ash pit or in bottom of producer. in. of water
48. Barometric pressure of atmosphere (as observed).....in. of mercury at ...°F.
49. Corresponding barometric pressure of atmosphere, in inches of mercury at 32°F.
50. Corresponding barometric pressure of atmosphere, in inches of mercury at 68°F.

Miscellaneous

51. Relative humidity of air.....per cent
52. Humidity of steam as delivered to the producer.....per cent wet
53. Average depth of fuel bed.....in.
54. Average depth of ash bed.....in.
55. Number of cleanings.....
56. Average length of cleanings.....hr.
57. Average intervals between cleanings.....hr.
58. Average intervals between pokings.....hr.
59. Average length of pokings.....min.
60. Ashes, clinkers, or refuse, wet or dry.....lb.

Temperatures

61. Temperature of feed water entering vaporizer of producer.....°F.
62. Temperature of feed water entering outside steam generator or waste heat boiler.....°F.
63. Temperature of air in room.....°F.
64. Temperature of air entering producer.....°F.
65. Temperature of gas leaving producer.....°F.
66. Temperature of tar and soot as delivered.....°F.
67. Temperature of gas in main at point where gas is measured.....°F.
68. Temperature of gas entering scrubber.....°F.
69. Temperature of gas leaving scrubber.....°F.
70. Temperature of gas entering purifier.....°F.
71. Temperature of gas leaving purifier.....°F.
72. Temperature of water entering scrubber.....°F.
73. Temperature of water leaving scrubber.....°F.
74. Temperature of water entering purifier.....°F.
75. Temperature of water leaving purifier.....°F.
76. Temperature of cooling water or seal water entering producer.....°F.
77. Temperature of cooling water leaving producer.....°F.
78. Temperature of ashes when being removed.....°F.
79. Temperature of
 - (a) Steam delivered to producer.....°F.
 - (b) Air-steam mixture in ash pit or bottom of producer.....°F.

Proximate Analysis of Fuel Supplied to Producer

80. Moisture.....per cent
81. Volatile matter and its nature.....per cent

82. Fixed carbon.....	per cent
83. Ash.....	per cent
84. Sulphur, separately determined.....	per cent
	100 per cent

Ultimate Analysis of Fuel

	As Fired	Dry Fuel
85. Carbon, C.....	per cent	per cent
86. Hydrogen, H ₂	per cent	per cent
87. Oxygen, O ₂	per cent	per cent
88. Nitrogen, N ₂	per cent	per cent
89. Sulphur, S.....	per cent	per cent
90. Ash.....	per cent	per cent
	100 per cent	

Analysis of Ashes and Refuse

91. Carbon.....	per cent
92. Inorganic earthy matter and iron.....	per cent
93. Moisture.....	per cent
94. Nature and texture of ashes.....	
95. Fusing temperature of ashes.....	°F.

Analysis of Dry Gas by Volume

96. Carbon dioxide, CO ₂	per cent
97. Carbon monoxide, CO.....	per cent
98. Oxygen, O ₂	per cent
99. Hydrogen, H ₂	per cent
100. Methane, or marsh gas, CH ₄	per cent
101. Ethylene, or olefiant gas, C ₂ H ₄	per cent
102. Hydrogen sulphide, H ₂ S.....	per cent
103. Sulphur dioxide, SO ₂	per cent
104. Ammonia, NH ₃	per cent
105. Nitrogen, N ₂ (by difference).....	per cent
106. Total combustible in dry gas.....	per cent

Calorific (Heating) Values by Calorimeter

107. Higher heating value of 1 lb. of fuel as fired.....	B.t.u.
108. Higher calorific value of 1 lb. of dry fuel.....	B.t.u.
109. Higher calorific value of 1 lb. of combustible by analysis.....	B.t.u.
110. Higher calorific value of 1 cu. ft. of dry gas at 68°F. and 29.92 in. of mercury.....	B.t.u.
111. Lower calorific value of 1 cu. ft. of dry gas at 68°F. and 29.92 in. of mercury.....	B.t.u.

Total and Hourly Quantities and Rates

Total Quantities

112. Gross weight of fuel charged corrected for estimated differences in weights of fuel in producer at beginning and end of test.....	lb.
113. Moisture in fuel gasified.....	per cent
114. Weight of dry fuel charged into producer or net weight of dry fuel.....	lb.
115. Dry fuel gasified:	
(a) Per sq. ft. of main fuel bed (use Item 34).....	lb.
(b) Per hr.....	lb.

116. Gross weight of fuel as fired used by auxiliary boiler for supplying steam to producer and its blower. lb.
117. Gross weight of fuel as fired used by auxiliary boiler for supplying steam to the pumps which supply water to producer, scrubber, and purifier. lb.
118. Total gross weight of fuel as fired supplied to plant. lb.
119. Total weight of dry fuel supplied to plant. lb.
120. Weight of wet ashes and refuse:
 - (a) From producer. lb.
 - (b) From auxiliary boilers. lb.
 - (c) From plant. lb.
121. Weight of dry ashes and refuse:
 - (a) From producer. lb.
 - (b) From auxiliary boilers. lb.
 - (c) From plant. lb.
122. Dry ashes and refuse to dry fuel:
 - (a) From producer. per cent
 - (b) From auxiliary boilers. per cent
 - (c) From plant. per cent
123. Weight of gross combustible charged:
 - (a) Into producer. lb.
 - (b) Into auxiliary boilers. lb.
 - (c) Into plant. lb.
124. Weight of unburnt carbon in 1 lb. of dry ashes and refuse:
 - (a) From producer. lb.
 - (b) From auxiliary boilers. lb.
 - (c) From plant. lb.
125. Weight of net combustible charged in producer:
 - (a) Into producer. lb.
 - (b) Into auxiliary boilers. lb.
 - (c) Into plant. lb.
126. Volume of air entering producer at . . . °F. cu. ft.
127. Its relative humidity. per cent
128. Weight of dry air entering producer. lb.
129. Volume of gross gas delivered at . . . °F. and . . . in. water pressure (or suction). cu. ft.
130. Specific gravity of dry gas.
131. Weight of dry gas delivered at . . . °F. and . . . in. water pressure (or suction) lb.
132. Moisture in gas leaving producer, mixed with 1 lb. of dry gas. lb.
133. Moisture in gas leaving scrubber, mixed with 1 lb. of dry gas. lb.
134. Equivalent volume of gross dry gas at temperature of 68°F. and pressure 29.92 in. of mercury (at 32°F.). cu. ft.
135. Equivalent volume of gross dry gas at 68°F and 29.92 in. of mercury (at 32°F.)
 - (a) Per lb. of dry fuel. cu. ft.
 - (b) Per lb. of net combustible. cu. ft.
136. Equivalent volume of net dry gas at 68°F. and 29.92 in. of mercury (at 32°F.)
 - (a) Per pound of dry fuel. cu. ft.
 - (b) Per pound of net combustible. cu. ft.
137. Weight of tar and soot delivered. lb.
138. Percentage of tar and soot in gas referred to dry fuel. per cent
139. Residual tar and soot in clean dry gas. grains per cu. ft.
140. Heat supplied to producer as steam
 - (a) From vaporizer. B.t.u.

(b) From waste-heat boiler.....	B.t.u.
(c) From separately fired boiler.....	B.t.u.
141. Water supplied to ashpit and water seal for vaporization.....	lb.
142. Water supplied to ashpit and water seal for cooling.....	lb.
143. Water supplied to vaporizer of producer or to boiler supplying steam to producer.....	lb.
144. Water supplied as moisture in fuel.....	lb.
145. Water supplied as moisture in air.....	lb.
146. Steam supplied to air blower and to producer.....	lb.
147. Total water and steam supplied to producer for gasification.....	lb.
148. Water supplied to producer and auxiliaries for cooling.....	lb.
149. Water supplied to scrubber.....	lb.
150. Water supplied to purifier.....	lb.
151. Water supplied to auxiliary boiler to generate steam for steam pumps.....	lb.
152. Water supplied to waste-heat boiler.....	lb.
153. Water supplied for other auxiliaries.....	lb.
154. Total weight of water supplied to producer and its auxiliaries.....	lb.

Efficiencies

155. Gross efficiency of producer, based on fuel as fired and steam supplied.....	per cent
156. Net efficiency of producer, based on fuel as fired and steam supplied.....	per cent
157. Gross efficiency of producer, based on dry fuel and steam supplied.....	per cent
158. Net efficiency of producer based on dry fuel and steam supplied.....	per cent
159. Gross efficiency of producer, based on net combustible and steam supplied.....	per cent
160. Net efficiency of producer, based on net combustible and steam supplied.....	per cent

Heat Balance Based on 1 Pound of Dry Fuel and the Products Therefrom

161. Higher calorific (heating) value of 1 lb. of dry fuel.....	B.t.u., = 100 per cent
162. Higher calorific (heating) value of dry gas per pound of fuel.....	B.t.u., = ... per cent
163. Sensible heat in hot dry gas above 68°F.....	B.t.u., = ... per cent
164. Total heat of moisture in gas above 68°F.....	B.t.u., = ... per cent
165. Heat removed by water of scrubber.....	B.t.u., = ... per cent
166. Heat removed by purifier.....	B.t.u., = ... per cent
167. Heat lost in residual tar and soot of dry gas.....	B.t.u., = ... per cent
168. Heat lost as unburnt fuel in ash.....	B.t.u., = ... per cent
169. Heat lost by radiation and otherwise unaccounted for.....	B.t.u., = ... per cent
170. Sensible heat of fuel above or below 68°F.....	B.t.u., = ... per cent
171. Sensible heat of air above or below 68°F.....	B.t.u., = ... per cent
172. Total heat above 68°F. in water and steam supplied.....	B.t.u., = ... per cent
173. Sensible heat of ashes and refuse.....	B.t.u., = ... per cent
174. Heat lost to cooling water in producer and auxiliaries.....	B.t.u., = ... per cent
175. Power required for supplying air, as by turboblower.....	hp. or kw.
176. Water pumps.....	hp. or kw.
177. Tar pumps.....	hp. or kw.
178. Scrubbers.....	hp. or kw.
179. Purifiers.....	hp. or kw.
180. Poking fire.....	hp. or kw.
181. Rotating producer.....	hp. or kw.
182. Exhauster.....	hp. or kw.
183. For charging producer.....	hp. or kw.

184. Total power required, mechanical or electrical.....hp. or kw.

When steam is supplied to producer from an outside source, the data and results should be modified accordingly.

Manual Labor Supplied and the Cost Thereof

During.....hours			
	Men-hr.	Rate of pay, cts. per hour	Cost, dollars
185. Generating steam:			
(a) For producer and its blower.....			
(b) For pumps.....			
186. Charging producer.....			
187. Poking fire.....			
188. Rotating producer.....			
189. Cleaning producer.....			
190. Operating auxiliaries.....			
191. Recovering unburnt fuel.....			
192. Other operations, such as removal of ashes..			
193. Total labor supplied for operation of plant....			

Gross and Net Costs, Exclusive of Overhead

	Per 1,000 cu. ft. net gas at 68°F., 29.92 in. at 32°F.	Per 1,000,000 B.t.u. of available heat energy in	
		Net gas at 68°F., 29.92 in. at 32°F.	Hot gas at°F.
194. Fuel as fired at \$... per ton.....			
195. Fuel as fired at \$... per gal.....			
196. Water at \$... per 1,000 cu. ft.....			
197. High pressure steam at.....			
198. Exhaust steam at.....			
199. Electricity, a.c. or d.c. at.....			
200. Total labor supplied.....			
201. Total gross cost of labor, fuel, water, steam, and current used.....			
202. Total gross cost of labor, fuel, water, steam, and current used per ton.....			
(2,000 lb.) or per gal. of fuel consumed.....dollars			

Credits

203. High pressure steam at.....
204. Low pressure steam at.....
205. Tar and soot at.....
206. Ammoniacal liquor (sp. gr. =.....) at.....
207. Total credits.....

208. Total net cost of labor, fuel, water,
steam, and current used.....
209. Total net cost of labor, fuel, water,
steam, and current used per ton
(2,000 lb.), or per gallon of fuel consumed dollars

Summary

210. Total capacity of plant, in tons (2,000 lb.) or gallons of fuel consumed in 24 hr.
211. Net heat output, in millions of B.t.u., delivered in 24 hr.....B.t.u.
212. Total net cost of generating same, exclusive of overhead charges.....dollars

NOTE.—Record the rates for No. 194, per ton of 2,000 pounds; for No. 195, per gallon; for No. 196, per 1,000 cubic feet; for No. 197, per 1,000 pounds; for No. 198, per 1,000 pounds; for No. 199, per kilowatt-hour; for No. 203, per 1,000 pounds; for No. 204, per 1,000 pounds; for No. 205, per 1,000 pounds; for No. 206, per 1,000 cubic feet.

CHAPTER XVII

TESTING OF VENTILATING FANS OR BLOWERS AND AIR COMPRESSORS

Centrifugal fans are used almost exclusively when large volumes of air are to be handled at a comparatively small pressure. Such a fan consists essentially of a number of plates, either flat or curved, attached to radial arms springing from a central hub through which the driving shaft passes, as in the "spider" type shown in Fig. 281, or the blades may be attached to a conical plate as in Fig. 282. Fans resembling either of these two designs are known commercially as the straight-blade type.

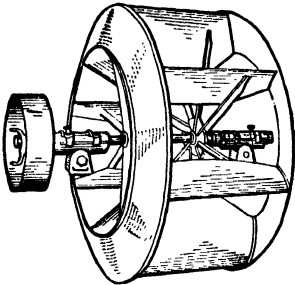


FIG. 281.—Spider-type fan.

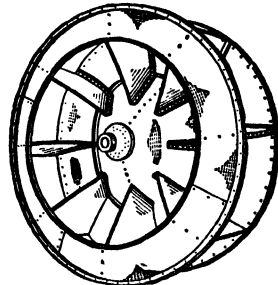


FIG. 282.—Conical-plate-type fan.

Disk or propeller fans are best illustrated by the so-called "electric" fans so commonly used in offices, shops, and dwellings. Fans of this type are usually of a very light construction with the vanes arranged as in a screw propeller for a ship. In many cases fans of this type are not provided with casings, so that it is more difficult to make velocity measurements than with other centrifugal fans.

Multiblade or turbine fans have an **impeller** or fan wheel of the "squirrel-cage" type, as illustrated in Fig. 283. Fans of this type can be designed to give very high efficiencies, due primarily to two characteristic features adopted in these designs. By the use of very short blades a very large intake space for the suction is provided which is practically unobstructed, thus giving a very free "suction." The other important feature of this fan is that the air leaves the blades at a higher velocity than that at which the tips of the blades are moving. The importance of this result is shown by a comparison of Figs. 284 and 285. The former illustrates the type of blading in a multiblade or turbine fan and shows graphically by a velocity diagram, constructed like a

parallelogram of forces, the velocity of the tips of the blades V_b , the velocity of radial flow in the blades V_r , and the absolute velocity of the discharge V_a , which is the velocity of the air with respect to the stationary casing. It will be observed that, in Fig. 284, V_a is nearly

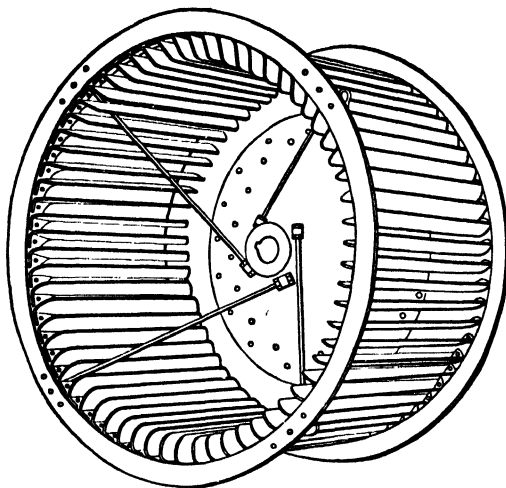


FIG. 283.—Turbine-type fan.

50 per cent greater than the velocity of the tips V_a , while in Fig. 285, representing the corresponding velocities for a straight-blade type of fan, the absolute velocity of the discharge V_a is considerably less than the speed of the tips of the blades. Increased velocity is accomplished in a type of fan like Fig. 284, not only by the curvature of the tips of the

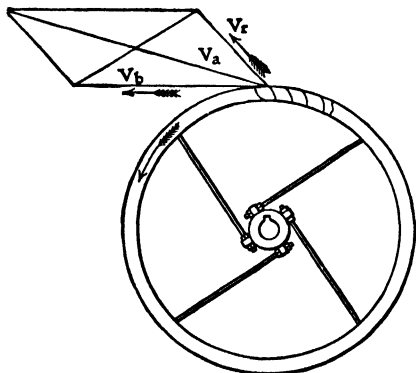


FIG. 284.—Velocity diagram for a turbine fan.

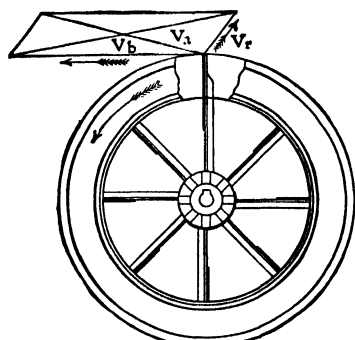


FIG. 285.—Velocity diagram for a straight-blade fan.

blades but also to some extent by making the blades somewhat concave with the inner ends (toward the center) practically radial. By this method of designing, the outer edges of the blades have a smaller space between them than the inner edges. This has the effect of reducing

the area on the discharge side of the blades, and consequently the velocity of the air is increased.

The work performed by a centrifugal type of fan is equal to the resistance times the velocity of flow. Since, however, the fan resistances are proportional to the square of the velocity,¹ the work done is proportional to the cube of the velocity.

Positive pressure blowers are used principally for blast furnaces and smelters where a higher pressure is needed than can be efficiently obtained with a centrifugal fan. A section showing the rotors and casing of one of these blowers is shown in Fig. 286. It is often called

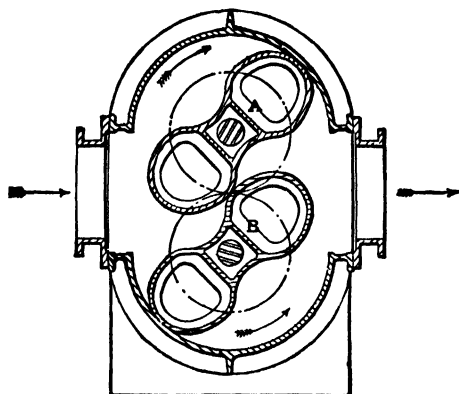


FIG. 286.—Typical positive pressure blower.

Root's blower. The efficiency of a blower of this type depends on the accuracy of the fitting of the two rotors *A* and *B*, with respect both to each other and to the casing. For this reason, when new, the efficiency is high, but after being in service for several years the bearings and the surfaces of the rotors will become worn, so that there is considerable leakage and consequent loss of efficiency.

Tests of ventilating fans or blowers are made frequently by a very simple method, that is, by determining the necessary data for calculating efficiency by measuring the work done by the fan "on the air" in giving velocity, and the power required to drive the fan alone, excluding bearing friction. The fan is preferably operated by an electric motor, of which the efficiency can be readily determined by a Prony brake test. The power required to overcome the bearing friction of the shaft of the fan may well be first determined by measuring the power input² (kilowatts) for a series of speeds when the keys fastening the fan to its shaft have been removed and the fan itself has been "blocked" in its casing or, still better, has been removed from the shaft. After the fan is attached

¹ See Professor Rateau's articles in *Rev. Mécanique*, Vol. 1, pp. 629-837.

² For normal operation "friction work" is, for machinery in general, proportional to the speed.

again to the shaft, the input to the motor and the work done by the fan "on the air" should be determined for various speeds. Then, obviously, the ratio of the work done by the fan divided by the power required to drive it after correction for the efficiency¹ of the motor and bearing friction is the actual efficiency of the fan. In general terms this may be stated as follows:

f = input to motor to drive motor and shaft of fan in bearings, in kilowatts.

i = input to motor to drive motor and fan, in kilowatts.

e = efficiency of motor for motor input of i kilowatts and at speed of test.

e' = efficiency of motor for motor input of f kilowatts and at speed of test.

Then if i_n is the net work in **horsepower** to drive fan alone,

$$i_n = \frac{ei - e'f}{0.746}. \quad (93)$$

If the fan to be tested is direct-connected to a steam engine, the test is usually made by measuring the indicated horsepower of the engine at the various speeds and also with the fan disconnected for no load. If it is possible to do so, the fan should be removed from the shaft for the no-load test to determine bearing friction.

The work done by the fan "on the air" is most readily calculated by the same method used to calculate the efficiency of hydraulic pumps (see page 567), that is, by multiplying the number of pounds of air delivered per unit of time by the head in feet of air corresponding to the discharge pressure. This product is obviously in terms of work in foot-pounds per unit of time, and, dividing by 33,000, the corresponding horsepower is obtained. By using the following symbols, the same results may be expressed, however, by the product of "pressure times volume" as follows:

v = velocity of air, in feet per second.

h = head, in feet of air necessary to produce a velocity of v feet per second, or

= water pressure p , in **inches** observed with a manometer, produced by the **velocity** of the air times the ratio of

$$\frac{\text{Weight of a cubic foot of water}}{\text{Weight of a cubic foot of air}^2}.$$

¹ Motor efficiency must be necessarily determined for the conditions of each test, that is, for the same kilowatts and speed as for each test.

² Weight of air taken for calculation must be that corresponding to the *total* pressure in the discharge pipe, the temperature, and the humidity. For tables of weight of air see pp. 187 and 444, also "Calculating and Testing Ventilating Systems," issued by U. S. Navy Department, Washington, D. C.

$$v = \sqrt{2gh} = \sqrt{2g \frac{p \times 62.3}{\text{weight cubic foot air for test}}} \quad (94)$$

and V_m = velocity, in feet per **minute**, is (taking $2g = 64.3$)

$$V_m = 1,096.2 \sqrt{\frac{p}{\text{weight cubic foot air for test}}} \quad (95)$$

Now the velocity in feet per minute V_m , multiplied by the area of the section at which the velocity was observed, gives cubic feet C of air discharged per minute, and if P is the total pressure (static + velocity) (p. 185) in pounds per square foot, then we have for j the “**air horsepower**” or the work done by the fan “on the air.”

$$j = \frac{CP}{33,000} \quad (96)$$

and efficiency of fan E is

$$E = \frac{j}{i_n} = \frac{CP}{33,000i_n} \quad (97)$$

Velocity measurements are usually made with a **Pitot tube**, consisting essentially as shown in Figs. 187 and 191 (page 184) of two tubes with openings at the end, arranged so that one of them “faces” in the direction of flow and the other extends in a radial direction. The former is subjected to the sum of the velocity and static pressures, while the latter receives only the static pressure.

Anemometers (page 188) are also frequently used for velocity measurements of air, but they are not generally so reliable as good Pitot tubes. Since, however, the observations can be taken directly in feet per minute, these instruments are used for nearly all work where no great accuracy is expected.

A psychrometric chart for determining relative humidity with a wet- and a dry-bulb thermometer (Fig. 288), or the sling psychrometer (Fig. 289) as prepared by Carrier, is given in Fig. 287.¹

The horizontal scale of the chart gives dry-bulb temperatures, and the inclined lines wet-bulb temperatures. The relative humidity is read directly from the intersection of these constant temperature lines. For example, if the dry-bulb temperature is 78°F. and the wet-bulb temperature is 65°F., the relative humidity is 0.50.

Relative humidity as read from Fig. 287 can be checked with the values given in Table XXIVA.

A sling psychrometer is much more accurate than the stationary wet- and dry-bulb type. It should be revolved at about 150 revolutions per minute.

¹ MOYER, JAMES A., and R. U. FRITZ, “Air Conditioning,” p. 9, McGraw-Hill Book Company, Inc., New York.

An example showing the method of calculation for j , the work done by the fan "on the air," may assist in making the method of calculation clearer.

TABLE XXIVA.—RELATIVE HUMIDITY, PER CENT

Dry-bulb temp., °F.	Difference between dry and wet thermometers, °F.							
	2	4	6	8	10	12	14	16
40	83	68	52	37	23	7		
50	87	74	61	49	38	27	16	5
60	89	78	68	58	48	39	30	21
70	90	81	72	64	55	48	40	33
80	91	83	75	68	61	54	47	41
90	92	85	78	71	65	58	52	47

A series of Pitot tube measurements taken at 10 different places in the cross section of an air duct shows that the "velocity" pressure was 0.795 and the total pressure 1.09 inches of water. The observations of barometric pressure and temperatures by wet- and dry-bulb thermom-

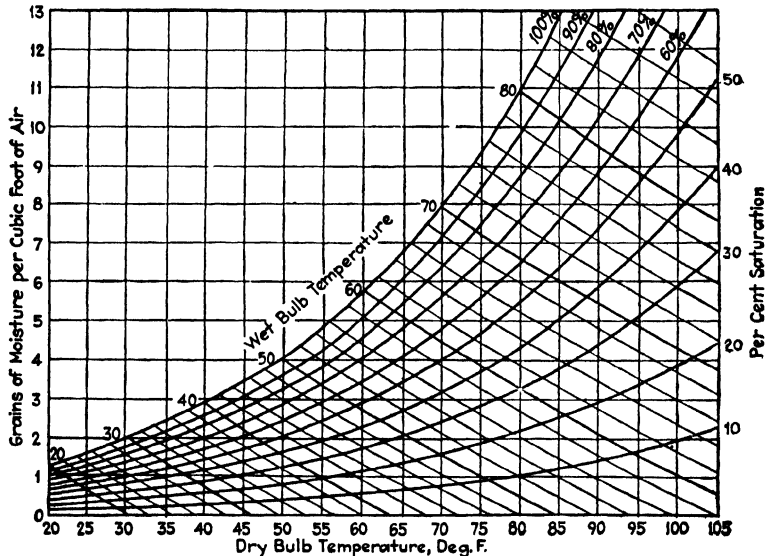


FIG. 287.—Psychrometric chart of relative humidity.

eters, together with the "total" pressure given above, served to determine the density or the weight of a cubic foot of air at the conditions of the test. Barometric pressure was 39.40 inches of mercury and temperatures of wet- and dry-bulb thermometers were, respectively, 54 and 71°F. According to the tables of the properties of air (see also

page 187) this was 0.07449 pound per cubic foot. Velocity of the air V_m in feet per minute is, therefore,

$$V_m = 1,096.2 \sqrt{\frac{0.795}{0.07449}} = 3,625 \text{ feet per minute.}$$

The diameter of the pipe was 10 inches, of which the area is 0.545 square foot. Cubic feet air discharged per minute (C) are $3,625 \times 0.545$, or 1,978. The total pressure P is the "total" pressure¹ in pounds per square foot or $\left(\frac{1.09}{13.6}\right) 0.491 \times 144$, or 5.66. Work done by the fan "on the air" is, then,

$$j = \frac{1,978 \times 5.66}{33,000} = 0.34 \text{ horsepower.}$$

Efficiency tests should be made with the fan operating under the discharge pressure for which it was designed or for which the guarantee was made, as the case may be. The efficiency of the fan may be considerably higher with a low discharge pressure than when connected up in a ventilating system where the discharge pressure is comparatively high.

Testing Ventilating Systems.—When tests are to be made of ventilating systems, precautions should be taken in the examination of all ducts and piping to see that they are clear of all lumber, rubbish, etc., and that the dampers are properly set. The tests consist usually in measuring in each system the "static" and the "total" pressures with a Pitot tube with all louvers open. These tests should be made with the fan running at high, low, and three or four intermediate speeds. All the results should be checked by plotting a curve with revolutions per minute for abscissas and cubic feet of air delivered per minute for ordinates. This curve should be approximately a straight line passing through the origin if all the louvers have remained open

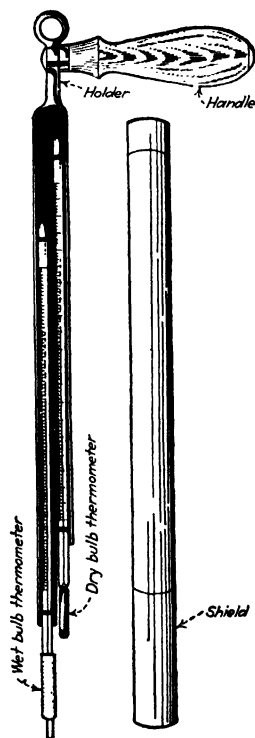


FIG. 289.—Sling psychrometer.

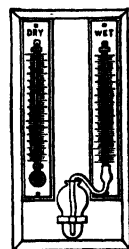


FIG. 288.—Wet- and dry-bulb thermometers.

throughout the tests. On the same abscissas, curves of pressure for ordinates will also be of advantage in showing the consistency of observa-

¹ In this expression 13.6 is the specific gravity of mercury, 0.491 is a factor for changing inches of mercury at room temperature to pounds per square inch, and 144 is used to change pounds per square inch to pounds per square foot.

tions. The location selected for the testing slot to be used for inserting the Pitot tube in the mains should be as near as possible to the fan; preferably no branches, however small, should run off between the fan and the testing slots in the mains. Furthermore, the testing slots should not be near turns and bends and particularly no turns or elbows should be immediately ahead of a slot, that is, in the direction toward the fan. These testing slots should be covered when not in use.

In the **U. S. Navy Department the standard conditions** adopted for testing ventilating fans and air-supply mains are a pressure of 5 pounds per square foot in the moving air at the discharge outlet from the fan and a velocity of 2,000 feet per minute. Air at the *standard conditions* is to be 70°F. and a relative humidity of 70 per cent. Under these standard conditions a cubic foot of air weighs 0.07488 pound. The pressure of 5 pounds is equivalent to a **pressure head** of 67 feet of standard density air. A velocity of 2,000 feet per minute corresponds to a **velocity head** of 17.27 feet. Total head against which the air is delivered to the supply mains for the standard conditions is then 84.27 feet, making a very satisfactory combination of velocity and pressure head, approaching as it does the maximum possible delivery for this pressure head.

Corrections for Losses of Total Head in Ducts.—There is always some loss of total head along a duct or pipe due to friction. As a result, there is a smaller delivery than that given for standard conditions. Using the following symbols:

h_f = loss of head, in feet, due to friction.

f = coefficient of friction = 0.00008 in piping of good construction.

l = length of duct, in feet.

d = diameter of duct, in feet.

V_m = velocity of flow through duct, in feet per minute.

$$h_f = \frac{V_m^2 l}{11,250,000 d} \quad (98)$$

If $V_m = 2,000$ then $h_f = 0.3556 \frac{l}{d}$.

Loss of head in a **square duct** is usually about the same as for a round one; but for a duct of **rectangular section** of which the short side is b and the long side is nb , the formula above becomes, with l and V_m as before,

$$h_f = \frac{1+n}{n} \times \frac{1}{b} \times \frac{V_m^2}{22,500,000} \quad (99)$$

With the help of these formulas when the size of the main ducts and the discharge in cubic feet per minute at each outlet are known, the head at each outlet as compared with the standard total of 84.27 feet can be calculated. As a "rough-and-ready" rule it is often stated that for

a loss of 1 foot of head there is a loss of 0.6 per cent in delivery (cubic feet per minute).

Fan Equations.—When applying the following fan equations to changing volume, pressure, and horsepower conditions, it is well to remember that, for all practical purposes in the range used, a fan whether disk or centrifugal may be considered a constant displacement medium; that is, against any given air resistance, for every revolution made, the fan moves the same quantity of air. Important fan equations are as follows, being intended, of course, for the same fan when operating against the same air resistance with assumed constant efficiency.

If V_1 , SP_1 , and HP_1 represent cubic feet of air per minute, static pressure, and horsepower at known conditions; and V_2 , SP_2 , and HP_2 are these quantities at the desired conditions, then for centrifugal fans of different sizes but of exactly the same proportions, if D is the diameter of the fan wheel in inches, the following equations may be stated:

$$\frac{V_2}{V_1} = \frac{RPM_2}{RPM_1} \quad (a)$$

$$\frac{SP_2}{SP_1} = \left(\frac{RPM_2}{RPM_1} \right)^2 \quad (b)$$

$$\frac{HP_2}{HP_1} = \left(\frac{RPM_2}{RPM_1} \right)^3 \quad (c)$$

$$\frac{V_2}{V_1} = \left(\frac{D_2}{D_1} \right)^3 \quad (d)$$

$$\frac{SP_2}{SP_1} = \left(\frac{D_2}{D_1} \right)^2 \quad (e)$$

$$\frac{HP_2}{HP_1} = \left(\frac{D_2}{D_1} \right)^5 \quad (f)$$

STANDARD TEST CODE FOR CENTRIFUGAL FANS

Adopted by the National Association of Fan Manufacturers and the American Society of Heating and Ventilating Engineers, 1922

Test Duct.—All centrifugal fans shall be tested with a straight discharge duct of uniform area equal to the fan discharge, and not less than 10 diameters in length. Readings shall be taken with a Pitot tube (see page 184) over a plane at right angles to the axis of the duct, with the impact end pointed upstream. These readings shall be taken three-quarters of the length of the test duct from the fan end. The outer end of the test duct shall be equipped with an adjustable device for symmetrically throttling the air.

Number of Readings.—In round ducts, not less than 20 readings shall be taken (not less than two traverses of 10 readings each along diameters at right angles to each other) at the centers of areas of five

equal-area concentric zones across the section of the duct. The distances of these centers from the center of the duct in terms of the radius, r , are $0.316r$, $0.548r$, $0.707r$, $0.837r$ and $0.949r$.

In *rectangular ducts*, the readings shall be taken in the center of equal areas over the cross section of the duct. The number of equal spaces shall not be less than 16 and need not be more than 64 in any case. When less than 64 readings are taken, the centers of these rectangular areas shall not be more than 6 inches apart.

Angle of Transformation Piece.—The greatest angle between the axis of the test duct and any longitudinal element shall not exceed 7 degrees.

Pressure-measuring Instruments.—The standard instruments for measuring pressures shall be the Pitot tube and the manometer. The Pitot tube shall have not less than *four* static orifices not exceeding 0.02 inch diameter. No such orifice shall be located at less than eight tube diameters from the upstream end of the tube and an equal distance from the elbow. The manometers for measuring static pressure and velocity pressure are to be filled with a light liquid—kerosene, gasoline, or alcohol—and tested in position by comparison with a water-filled U-tube gage. This calibration shall be made with the rubber tubing attached, and the rubber tubing shall not be removed from the manometer after calibration. The manometer for measuring total pressure shall be vertical and water filled. All instruments shall be calibrated immediately before and after the test.

Dynamometer.—The cradle dynamometer is recommended for all power measurements. Direct-connected motors carefully calibrated may be used. Where indirect drive is used, friction tests shall be made to determine losses for each of the conditions of the tests.

Inlet or Outlet Obstructions.—Fan performances are based on unobstructed inlets and outlets except for the structural features of the fan involved in bearings and bearing supports. If a fan is provided with bearings, it shall be tested on its own shaft and in its own bearings and shall be run for several hours to wear it.

Number of Determinations.—Eight or more determinations covering a range of performance from approximately free delivery to no delivery shall be made with approximately equal increments of capacity or discharge area.

Observations and Calculations.—Simultaneous readings of total pressure, static pressure, velocity pressure, horsepower input, and speed shall be taken. Barometric pressure and wet- and dry-bulb temperatures shall be taken during each determination. Temperatures shall be taken both in the room and within the discharge duct as close as practicable to the point of traverse.

Air readings shall be corrected to standard conditions (page 444) at the fan inlet, that is, 0.07488 pound per cubic foot. The pressures and

horsepower input vary directly as the weight of air at constant speed. The velocities corresponding to the velocity pressure readings shall be averaged to obtain the average velocity for each traverse. The velocity is determined from the velocity pressure by the formula $V = 1096.2\sqrt{P/w}$ feet per minute, where P is the velocity pressure in inches of water and w is the weight of the air in pounds per cubic foot.

Capacity, static pressure, total pressure, and horsepower input shall be corrected to constant speed. For speed corrections, capacity varies directly as the speed, pressures as the square of the speed, and horsepower as the speed cubed for constant-discharge conditions. The total pressure and the static pressure shall also be corrected for duct friction, to give conditions at fan discharge, by adding the loss of head due to friction of test duct between the fan outlet and Pitot tube. This loss will be $f = 0.0257LP/D$ for round ducts and $f = 0.01285L(a + b)P/ab$ for rectangular ducts, where f = pressure loss in inches of water, P = average velocity pressure in inches of water, L = distance from fan outlet to Pitot tube, in feet, D = diameter of discharge duct, in feet, a and b = lengths of sides of duct, in feet.

The results, as corrected for standard air conditions and constant speed, shall be plotted, using capacity as abscissas, and pressures and horsepowers as ordinates. Smooth curves are then drawn through these points giving pressure and horsepower characteristics. From these data the mechanical efficiency is calculated. The mechanical efficiency is then plotted against capacity. These curves form the basis from which fan performance tables and characteristic charts are computed.

In calculating the performance of fans of the same design and similar proportions, but of other sizes and at other speeds, the following law applies: Up to and within $1\frac{1}{2}$ pounds per square inch pressure difference, at the same peripheral speed and discharge condition the delivery varies as the square of the diameter of the wheel, or for different speeds the delivery varies as the cube of the diameter times the number of revolutions.

Testing Exhaust Fans.—The capacity shall be determined as above. The inlet duct shall have an area equal to the fan inlet and six diameters in length. The total head is the difference between the average absolute total pressure in the discharge duct and the inlet duct with corrections as above for friction between the points of measurement and the fan. Readings in the inlet duct shall be taken at a place three-quarters of the inlet duct length from the entering end.

Tolerances.—A variation of $2\frac{1}{2}$ per cent in the determinations of capacity, pressure, or horsepower will not be considered excessive.

Tests of Disk and Propeller Fans.—Tests of these fans shall be made when the fans are exhausting from a chamber, and all air readings must be taken on the inlet side. The test set-up shall consist of an airtight cham-

ber of rectangular or drum form. The length of this chamber shall be $3d$, the diameter or length of side $2\frac{1}{2}d$, where d is the fan diameter.

At the entrance side of the chamber shall be a duct that is 10 duct diameters in length; this length includes a transformation piece connecting to the chamber. The transformation piece shall be carried to the full dimensions of the end of the chamber and provided with an effective diffuser.

The volumes handled shall be varied by restriction at the inlet end of the duct and free delivery conditions shall be procured by the use of an

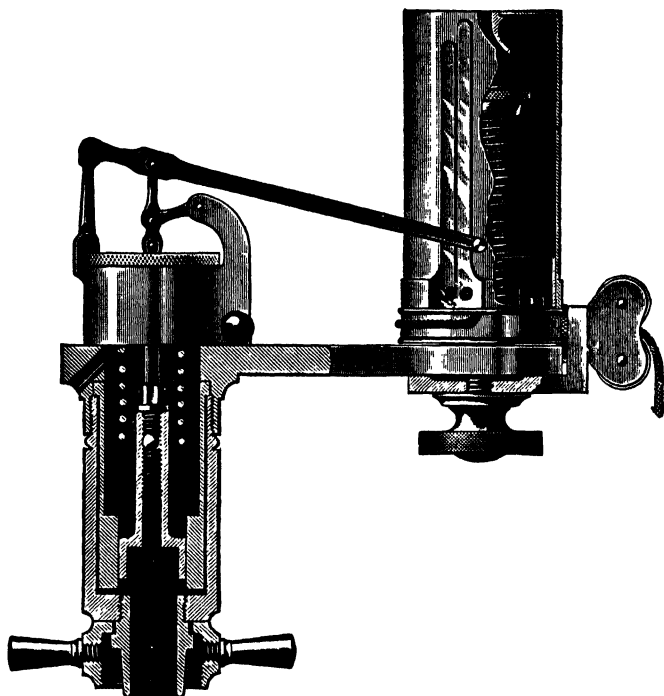


FIG. 290.—Crosby high-pressure indicator (ordnance type).

auxiliary blower for obtaining zero or above static pressures in the chamber. The capacity of the fan shall be determined as for centrifugal fans. The static pressure shall be determined by a reading of an impact tube in the large chamber.

Disk fans for blowing must have on the discharge side of the fan a collar of the same diameter as the fan opening and one diameter long.

Mounting.—Any obstruction to the flow of air that forms part of the unit shall be in its designed position.

Testing Air Compressors.—The various types of machines for compressing air are usually operated either by a steam engine or by an electric motor. Power delivered to the compressors by the engine or

motor is therefore measured by one of the methods already outlined for ventilating fans. Air compressors, particularly of the reciprocating type, are designed, as a rule, for operation at considerably higher pressures than would be suitable for ventilating fans, and the volume delivered must usually be measured in a comparatively small pipe. Air at high pressure is generally measured by calculating the flow through an orifice in a receiver or one of the other methods described on pages 190 to 194.

Foot-pounds of work "done on the air" per second, and this product divided by the **net power** required to drive the compressor, is the "**net**" **mechanical efficiency** (see also page 433).

In an air compressor in which the air cylinder is direct-connected to the steam cylinder the net power required to drive the compressor is determined by finding the indicated horsepower of the air cylinder and adding the friction in the air cylinder, which in many cases can be assumed to be half the difference between the indicated horsepower as measured in the steam and air cylinders. For use on air compressors operating at high pressures special indicators are made. One of the best is made by the Crosby Steam Gage and Valve Co., Boston, Mass., and is illustrated in Fig. 290. It is similar in design to the gas-engine indicator illustrated in Fig. 279 (page 407), except that the piston in the lower cylinder of the indicator is very small, only $\frac{1}{16}$ square inch in area. This indicator can therefore be readily used for pressures as high as 10,000 pounds per square inch.

A.S.M.E. TEST CODE (1926) FOR DISPLACEMENT COMPRESSORS AND BLOWERS (ABRIDGED)

The Code for Displacement Compressors and Blowers is intended as a guide for testing reciprocating piston compressors and blowers, as well as rotary compressors and blowers of the positive displacement type. The tables for data and results apply to complete units, including compressor element and driving element, but the Code itself constitutes a set of rules for the test of the compressor element only. For the driving element the Codes for Steam Engines, Steam Turbines, Internal-combustion Engines, etc., depending on the method of drive, should be consulted and followed.

Measurements.—The measurements to be made in any test of a compressor or blower have for their purpose the determination of the following essential quantities:

(a) The quantity of air or gas compressed and delivered expressed in cubic feet in terms of gas or air under intake conditions of temperature and pressure; (b) the average intake and discharge pressures; (c) the power required to compress and deliver the measured amount of air or gas handled; (d) the amount of steam, electrical energy, or fuel, depending upon the method of drive, consumed by the driving element during the test.

In order to arrive at these quantities and as a means of interpreting them, the following measurements are necessary:

(*e*) diameter and stroke of compressor cylinders, or equivalent dimensions for positive rotary blower; (*f*) diameter of piston rods; (*g*) number and dimensions of intercoolers and aftercoolers; (*h*) principal dimensions of driving element (see appropriate code); (*i*) speed in revolutions per minute and total number of revolutions; (*j*) indicated horsepower in compressor and driving cylinders (steam engine, internal-combustion engine, etc.); (*k*) discharge pressure; (*l*) intake pressure; (*m*) intercooler pressure; (*n*) barometric pressure; (*o*) temperature of air or gas before and after each stage; (*p*) temperature of cooling water at inlet and outlet of each jacket and cooler; (*q*) quantity of jacket and cooling water; (*r*) temperatures, pressures, etc., steam, gas, or oil as applied to driving element (see appropriate code).

Instruments and apparatus necessary are:

(*a*) Gaging tank and nozzle or orifice for measuring air or gas compressed; (*b*) tanks and platform scales of suitable capacity for measuring the quantity of condensed steam when a steam engine constitutes the driving element or for measuring fuel oil when an oil engine constitutes the driving element; (*c*) water meters or calibrated tanks, or tanks and platform scales for measuring jacket and intercooler circulating water; (*d*) pressure gages, vacuum gages, water and mercury manometers, and thermometers; (*e*) barometer; (*f*) steam calorimeter (for steam-engine-driven compressor); (*g*) voltmeter, ammeter, wattmeter and power-factor meter for measuring electrical input for motor-driven compressors; (*h*) revolution counter; (*i*) indicators; (*j*) a planimeter; (*k*) a dead-weight gage tester.

The measurement of the quantity of air compressed shall be made by one of the following methods: (*a*) in the case of a compressor or blower—flow into the atmosphere through nozzle or orifice from a gaging tank on the discharge side of the compressor or blower; (*b*) in the case of a vacuum pump—flow from atmosphere through a nozzle or orifice into a gaging tank on the intake side of the vacuum pump.

Where the foregoing prescribed method is impracticable, as, for example, in the case of a compressor handling inflammable gas or gas which cannot be wasted, one of the following instruments may be used. These alternatives are listed in order of preference: (*c*) nozzle or orifice in discharge line; (*d*) receiver tanks of known capacity into which the discharge from the compressor may be delivered to be used only for compressors working at a discharge pressure of 1,000 pounds per square inch or greater.

If method (*c*) is used, the instrument should be applied in a pipe through which the gas is permitted to flow under a pressure considerably less than the compressor discharge pressure. The reduction in pressure can be brought about by throttling and will result in eliminating, or at least minimizing, the pulsations in flow induced by the compressor.

The kind and size of gaging tank and nozzle or orifices selected will depend on the particular method chosen for measuring the quantity of air or gas compressed. Whenever possible, a low-pressure nozzle or orifice shall be used. A low-pressure nozzle is one through which the drop in pressure is small, that is, one in which the upstream absolute pressure is less than twice the downstream absolute pressure. Its use for high- or moderate-pressure compressors involves throttling the gas before it is admitted to the gaging tank to such an extent that all, or nearly all, pulsation is eliminated. If a high-pressure nozzle is selected, that is, one through which the drop in pressure is relatively large ($p_1 > 2p_2$), care should be exercised to select an orifice of a size sufficient to insure as much throttling between the discharge receiver and the gaging tank as possible.

The measurement of the intake and discharge pressures shall be by gage attached at the desired point, the fluctuations of which are reduced as far as may be by choking the gage cock. When extreme accuracy is desired or when the fluctuations are large, the average pressure may be found by working up pipe diagrams taken at or near the same point, thereby determining the mean pressure for the entire stroke.

Dimensions.—The dimensions of the compressor cylinders should be determined by actually measuring the machine itself without reference to the drawings. If the cylinders are much worn, the average diameter should be taken, but proper record of the fact should be noted.

The method for measuring clearance will depend upon the particular design of the compressor in question. In a reciprocating displacement compressor with valves of the automatic type, the most satisfactory method will be to calculate the clearance from the measured dimensions of the cylinder, its several ports, and passages. If detailed drawings are available, it will be found convenient to check each dimension of the parts involved by actual measurement and from this information compute the clearance volume. If mechanically operated valves are used, or if the construction of the valves of the automatic type is such as to permit blocking them in their closed position, the clearance may be obtained by the water-measurement method. This should only be attempted, however, when extreme accuracy is necessary or desirable.

The area of the compressor cylinder jacket surface is that part of the cylinder wall or cylinder-head wall in contact with the water circulation and should be measured on the dry side of the metal.

The intercooler and aftercooler surface should be measured on the dry side of the cooling surface. Surface in contact with the air to be cooled on the one side and atmospheric air on the other side should not be included as cooling surface. If a record of its amount is made, it should be listed as a separate item.

Operating Conditions.—Tests to determine the performance under working conditions in which no attempt at uniformity is made are not advised, since the only available method of air measurement involves the determination of the rate of flow, and total quantities of gas or air must be computed from the observed rate. Unless all conditions of operation are kept uniform or nearly uniform, the results obtained will not be of any value.

Starting and Stopping.—The compressor should be operated under test conditions for a considerable period before a record of its performance is made. This period should be of sufficient duration to bring about a steady condition of pressure, temperature, and speed.

Duration.—The duration of compressor tests will generally be governed by the requirements in so far as the driving element is concerned, and for direction as to the necessary period reference to the appropriate code governing the driving element should be made. In the case of a motor-driven unit accurate results can be obtained from very short tests, the controlling factor being the time required to record enough sets of observations to demonstrate the uniformity of running conditions during the test. The minimum duration of any compressor test must not be less than 1 hour.

Instruments should be read and indicator cards taken from each end of each cylinder at least quarter hourly when the conditions are uniform,

and oftener when there is much variation. For short tests, readings and cards should be taken much more often. If there are wide fluctuations in readings, they should be shown by recording instruments. Each indicator card should be marked with the number, date, time, corrected scale of spring, and end of cylinder, and on one card of each set, the readings of the pressure gages should be recorded. The log should contain the readings of discharge, intercooler and intake gages and thermometers, revolution counter, speed indicator, gaging-tank pressures and temperatures, and all other instruments, and these readings should be obtained at practically the same time the indicator cards are taken. The areas, lengths, and mean effective pressures shown by the indicator cards should also be entered in the log. A set of specimen indicator diagrams should be carefully selected from the whole number taken, and these should be embodied in the record. The specimen cards selected should be such as to show the average conditions of pressure.

Volume of Air or Gas Compressed.—The measurement of the amount of air or gas through a nozzle, orifice, or venturi tube results in a determination expressed in pounds per unit of time. The results obtained must therefore be reduced to terms of volume at the temperature and pressure obtaining at the compressor or blower intake. For this purpose the required specific weight at intake conditions may be calculated from the following formula:

$$\text{Specific weight} = \frac{p \times 144}{RT}, \text{ in pounds per cubic foot,}$$

where p = absolute intake pressure, in pounds per square inch.

T = absolute intake temperature, in degrees Fahrenheit.

R = constant (= 53.3 for air).

This assumes that the fluid compressed is ideal gas, an assumption quite satisfactory for air. The weight of air or gas per minute as shown by nozzle measurement divided by the specific weight at *intake conditions* is equal to the volume of free gas or air compressed per minute.

When natural gas is compressed, the value for R is not nearly enough constant under all conditions to satisfy engineering requirements and a correction must be made, depending in amount on the ratio of compression and the nature of the gas.

Correction for Moisture.—In the case of a compressor compressing moist air, if some of the moisture is condensed and removed during the process of intercooling and aftercooling, the weight of the air and water-vapor mixture which passes through the measuring nozzle will be less than that taken in by the compressor. In reducing the amount of air as shown by the nozzle measurements to terms of air at intake pressure and temperature, it will be necessary to correct for the water thus removed, and this correction can most readily be made by reducing the capacity to terms of air at intake total pressure, intake temperature, and discharge specific humidity. This correction may be made in either of two ways—(1) through the **vapor pressure** or (2) through

the vapor density. Both methods should be used in order to check results. This is advisable because in any case the correction will be small.

Correction through Vapor Pressure.—Determine the pressure of saturated vapor corresponding to the dry-bulb temperature of the discharge air. Multiply this by the ratio of the absolute intake total pressure to the absolute discharge total pressure. The result will be the corrected intake vapor pressure. From the steam tables determine the pressure of saturated vapor at intake temperature. Multiply this by the relative humidity of air at intake conditions. This will give the actual intake vapor pressure. Subtract this from the total intake pressure (barometric) to obtain the partial pressure (page 247) of air actually in the intake. Add the corrected intake vapor pressure to the partial pressure of air actually prevailing in the intake. This will give the corrected intake total pressure. Substitute this pressure in the foregoing formula, and compute the specific weight. Divide this into the weight of dry air discharged to obtain the corrected intake volume by the first method.

Correction through Vapor Density.—Compute the volume of air compressed as directed above, using the intake total pressure. Determine the total weight of water drained from all coolers. Divide the water drained per unit of time by the volume of intake air for the same unit of time. This will give the water drained per cubic foot of intake air. From psychrometric tables or charts (page 442), find the vapor density corresponding to the wet- and dry-bulb thermometer readings in the compressor intake. From this density subtract the weight of water drained from the coolers per cubic foot of intake air to obtain the corrected intake vapor density. From steam tables determine the density of saturated vapor at intake temperature. Divide the corrected intake vapor density by the density of saturated vapor at intake temperature. This will give the corrected relative humidity. From the steam tables determine the pressure of saturated vapor at intake temperature. Multiply the pressure of saturated vapor by the corrected relative humidity. This will give the corrected intake vapor pressure. Add this to the partial pressure of air actually prevailing in the intake. This will give the corrected intake total pressure. Substitute this pressure in the foregoing formula and compute the specific weight. Divide this into the weight of dry air discharged to obtain the corrected intake volume by the second method.

It should be noted that the correction for moisture may properly be made only if moisture in the form of condensate is actually removed from the coolers. Care must be taken to make sure that the discharge temperature and pressure used in the computations are taken at or near the outlet of the cooler. If no aftercooler is installed, the discharge temperature and pressure at or near the outlet of the intercooler should be used in the calculation of the correction through vapor pressure, as outlined above. That is, the air is assumed to carry saturated vapor as it leaves the *last* cooler from which water is actually drained, and the air conditions must be observed immediately at the exit of that cooler.

Indicated Horsepower.—The indicated horsepower for each end of the cylinder is found by using the formula:

$$\text{i.hp.} = \frac{p_m l a n}{33,000},$$

where p_m = mean effective pressure, in pounds per square inch.

l = length of the stroke, in feet.

a = area in square inches of the piston less the area of the piston rod, if any.

n = number of revolutions per minute.

The total horsepower of the cylinder is the sum of the horsepower developed in the two ends.

The air horsepower required to compress adiabatically the measured quantity of air at the average rate for a single stage or for any one stage of multistage compressors will be

$$\text{hp.} = \frac{144k}{33,000(k-1)} p_a v_a \left[\left(\frac{p_b}{p_a} \right)^{\frac{k-1}{k}} - 1 \right],$$

where v_a = volume compressed, in cubic feet per minute; p_a = absolute intake pressure, in pounds per square inch; p_b = absolute discharge pressure, in pounds per square inch; k = ratio of specific heats (= 1.40 for air); N = number of stages in the compressor.

For air the foregoing formula becomes

$$\text{hp.} = 0.0153 p_a v_a \left[\left(\frac{p_b}{p_a} \right)^{0.29} - 1 \right].$$

For any particular cylinder in a multistage compressor substitute the absolute intake pressure at the cylinder in question for p_a , and absolute discharge pressure for the particular cylinder in question for p_b in the foregoing formula.

The minimum horsepower required to compress adiabatically the measured quantity of air at the average rate for a multistage compressor with perfect intercooling (identical intake temperature to each stage) will be

$$\text{hp.} = \frac{144kN}{33,000(k-1)} p_a v_a \left[\left(\frac{p_b}{p_a} \right)^{\frac{k-1}{Nk}} - 1 \right].$$

For air the foregoing formula becomes

$$\text{hp.} = 0.0153 N p_a v_a \left[\left(\frac{p_b}{p_a} \right)^{\frac{0.29}{N}} - 1 \right].$$

The air horsepower required to compress isothermally the measured quantity of air at the average rate will be

$$\text{hp.} = \frac{144}{33,000} p_a v_a \log_e \frac{p_b}{p_a}.$$

Gross Horsepower.—The gross horsepower is the indicated horsepower in the steam or power cylinders in the case of a steam or internal-combustion-engine-driven compressor, the electrical horsepower multiplied by the motor efficiency in the case of a motor-driven compressor, and the brake horsepower delivered to the compressor shaft in the case of a belt-driven compressor. In the case of a compressor driven through

a belt or gear, the power absorbed by the belt or gear, must be taken into account.

Electrical Horsepower.—The electrical horsepower of a motor is found by dividing the input at the terminals expressed in kilowatts by the constant 0.746. In the case of an alternating-current motor, the input determined, whether expressed in electrical horsepower or kilowatts, should be the total input. When the power for excitation or ventilation is taken directly from the compressor shaft, the total input is that indicated at the alternating-current motor terminals; but if the motor efficiency as given does not account for the exciter and fan input and exciter loss, the total electrical horsepower used in calculating the gross horsepower is that indicated at the motor terminals less an equivalent power required to run the exciter.

Volumetric Efficiency.—The volumetric efficiency is the ratio of the capacity of the compressor to displacement. The capacity is the actual amount of air or gas compressed and delivered, expressed in cubic feet per minute at intake temperature and pressure.

Mechanical Efficiency.—The mechanical efficiency is the ratio of the air indicated horsepower to the indicated horsepower in the power cylinders in the case of a steam-driven or internal-combustion-engine-driven compressor, and to the brake horsepower delivered to the shaft in the case of a power-driven machine.

Compression Efficiency.—The compression efficiency in a single-stage compressor is the ratio of the horsepower required to compress adiabatically all the air or gas delivered by the compressor to the horsepower developed in the compressor cylinder as shown by the indicator cards. The compression efficiency for any particular cylinder of a multistage compressor is the ratio of the horsepower required to compress adiabatically all the air or gas delivered by the compressor through the observed pressure range for the particular cylinder in question to the horsepower developed in the particular cylinder in question as shown by the indicator cards. The equivalent compression efficiency for all cylinders in a multistage compressor is the ratio of the minimum air horsepower required to compress adiabatically all the air or gas delivered by the compressor to the total horsepower developed in the compressor cylinders as shown by the indicator cards.

Overall Efficiency.—The overall efficiency is the ratio of the isothermal horsepower to the gross horsepower.

TABLE XXV.—DATA AND RESULTS OF TEST ON POWER-DRIVEN DISPLACEMENT AIR COMPRESSOR—COMPRESSING ELEMENT

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

Description, Dimensions, Etc., of Compressor¹

7. Type of compressor (single stage or multistage, and kind of gas).....			
8. Type of compressor valves.....			
9. Method of driving compressor.....			
10. Method of volume control.....			
11. Rated discharge pressure.....		lb. per sq. in.	
12. Rated speed.....		r.p.m.	
13. Rated displacement.....		cu. ft. per min.	
14. Rated output expressed in volume (air or gas at intake pressure and temperature).....		cu. ft. per min.	
15. Type of intercoolers and aftercoolers.....		First Inter- cooler	Second Inter- cooler
16. Area of water-cooled surface, square feet.....			
	First Cylinder	Second Cylinder	Third Cylinder
17. Diameter of compressor cylinders, inches....			
18. Stroke of pistons, feet.....			
19. Diameter of piston rods or tail rods, inches..			
20. Clearance in terms of piston displacement per cent			
(a) Clearance, head end.....			
(b) Clearance, crank end.....			
(c) Clearance, average.....			
21. Air-cylinder ratio based on piston displace- ment:			
(a) First stage to second stage.....			
(b) Second stage to third stage.....			
(c) Total (first stage to final stage).....			
22. Horsepower constant, air cylinder			
(a) Head end (stroke $\times$ net piston area $\div$ 33,000).....			
(b) Crank end (stroke $\times$ net piston area $\div$ 33,000).....			
23. Area of compressor cylinder-jacketed surface.....		sq. ft.	
24. Length and cross-sectional area, intake pipe.....		ft.	
25. Diameter of final discharge pipe.....		in.	

Test Data and Results

40. Duration of test.....hr.

Average Pressures

41. Barometer (as observed).....in. of mercury
 (a) Corresponding absolute pressure.....lb. per sq. in.
42. Pressure by gage at *intake* near cylinder.....lb. per sq. in.
 (a) Corresponding absolute pressure.....lb. per sq. in.

¹ For description, dimensions, etc., of compressor driving element see Table XVI or XVII (pp. 458 and 459).

- (b) Maximum pressure by indicator diagram.....lb. per sq. in.
 (c) Minimum pressure by indicator diagram.....lb. per sq. in.
 43. Pressure above atmosphere in *discharge* pipe near high-pressure
 cylinder by gage.....lb. per sq. in.
 (a) Corresponding absolute pressure.....lb. per sq. in.
 (b) Maximum pressure by indicator diagram.....lb. per sq. in.
 (c) Minimum pressure by indicator diagram.....lb. per sq. in.
 44. Pressure in first intercooler by gage.....lb. per sq. in.
 45. Pressure in second intercooler by gage.....lb. per sq. in.

Average Temperatures

50. Room temperature.....°F.
 51. Temperature of air or gas near intake, dry-bulb.....°F.
 52. Temperature of air or gas near intake, wet-bulb.....°F.
- | | First
Cylinder | Second
Cylinder | Third
Cylinder |
|---|-------------------|--------------------|-------------------|
| 53. Temperature of air or gas in suction port, °F.. | | | |
| 54. Temperature of air or gas in discharge port, °F. | | | |
| 55. Temperature of jacket cooling-water inlet, °F. | | | |
| 56. Temperature of jacket cooling-water outlet,
°F..... | | | |
| | | First
Cooler | Second
Cooler |
| 57. Temperature of cooler circulating-water inlet, °F..... | | | |
| 58. Temperature of cooler circulating-water outlet, °F..... | | | |
| 59. Temperature of air or gas in gaging tank at upstream
side of orifice °F..... | | | |

Total Quantities

65. Relative humidity of air or gas at intake of compressor (use Smithsonian
 Tables)¹.....per cent
 66. Air compressed and delivered to line.....lb.
 67. Air or gas compressed and delivered to line expressed in cubic feet at
 intake pressure and temperature.....cu. ft.

- | | First
Cylinder | Second
Cylinder | Third
Cylinder |
|---|-------------------|--------------------|-------------------|
| 68. Cooling water supplied jackets, pounds..... | | | |
| | | First
Cooler | Second
Cooler |
| 69. Cooling water supplied coolers, pounds..... | | | |

Unit Quantities

76. Volume of air or gas at intake pressure and temperature compressed and
 delivered per minute [Item 67 ÷ (60 × Item 40)].....cu. ft.
- | | First
Cylinder | Second
Cylinder | Third
Cylinder |
|--|-------------------|--------------------|-------------------|
| 77. Cooling water supplied jackets, pounds or
gallons per minute [Item 68 ÷ (60 ×
Item 40)]..... | | | |

¹ More recent data are given in the *psychrometric chart*, page 442. This chart is from Moyer and Fittz, "Air Conditioning," McGraw-Hill Book Company, Inc., New York, 1933.

	First Stage	Second Stage	Third Stage
78. Cooling water supplied intercoolers, pounds or gallons per minute [Item 69 $\div$ (60 $\times$ Item 40)].....			
Speed			
82. Total number of revolutions as shown by compressor counter.....			
83. Revolutions per minute [Item 82 $\div$ (60 $\times$ Item 40)].....			r.p.m.
84. Average piston speed.....			ft. per min.
Power			
85. Indicated horsepower of compressor cylinders, whole compressor.....			i.hp.
86. Low-pressure compressor cylinder, horsepower:			
Crank end.....			i.hp.
Head end.....			i.hp.
87. Intermediate-pressure compressor cylinder, horsepower:			
Crank end.....			i.hp.
Head end.....			i.hp.
88. High-pressure compressor cylinder, horsepower:			
Crank end.....			i.hp.
Head end.....			i.hp.
89. Friction horsepower [Item 100 (Table XVI, p. 459) or Item 90 (Table XVII, p. 460)—Item 85].....			i.hp.

Power Input

See Tables XVI and XVII.

Economy Results

101. Gross horsepower per 100 cu. ft. of air or gas delivered per minute at intake pressure and temperature (p. 454).....	hp.
102. Indicated horsepower in compressor cylinder per 100 cu. ft. of air or gas delivered per minute at intake pressure and temperature (Item 85 $\div$ Item 76).....	hp.

Efficiency Results

108. Horsepower required to compress adiabatically measured quantity of air or gas at average speed of compressor during test (p. 454).....	hp.
109. Minimum horsepower required to compress adiabatically measured quantity of air or gas at average speed of compressor during test (p. 454).....	hp.
110. Horsepower required to compress isothermally measured quantity of air or gas at average speed of compressor during test.....	hp.
111. Volumetric efficiency (p. 455).....	per cent
112. Compression efficiency (Item 108 $\div$ Item 86, 87, or 88).....	per cent
113. Equivalent compression efficiency all cylinders (Item 109 $\div$ Item 85).....	per cent
114. Mechanical efficiency.....	per cent
115. Overall efficiency (Item 110 $\div$ gross hp.).....	per cent

**TABLE XXVI.—ADDITIONAL ITEMS APPLYING ONLY WHEN THE DRIVING ELEMENT
IS AN ELECTRIC MOTOR***Description, Dimensions, Etc., of Compressor Drive*

26. Type of motor.....	
27. Rated power of motor.....	

28. Volts.....	
29. Amperes.....	
30. Phase.....	
31. Cycles.....	
32. Revolutions per minute.....	
33. Type and rating of exciter.....	
(a) Volts.....	
(b) Amperes.....	

Power Input

90. Volts.....	
91. Amperes per phase.....	
92. Power factor (from meter).....	
93. Exciter output:	
(a) Volts.....	
(b) Amperes.....	
94. Efficiency of exciter (from builder's test including belt if any).....	per cent
95. Horsepower input to motor by wattmeter.....	hp.
96. Efficiency of motor (from builder's test).....	hp.
97. Net motor horsepower.....	hp.
98. Horsepower output for exciter.....	hp.
99. Net horsepower to drive exciter.....	hp.
100. Horsepower to drive compressor or blower.....	hp.
103. Electrical horsepower per 100 cu. ft. of air or gas delivered per minute at intake pressure and temperature (p. 455).....	hp.

TABLE XXVII.—ADDITIONAL ITEMS APPLYING ONLY WHEN THE DRIVING ELEMENT
IS A STEAM ENGINE

Description, Dimensions, Etc., of Compressor Drive

26. Type of engine (simple or multiple expansion).....	
27. Type of steam valves.....	
28. Auxiliaries (steam or electric drive).....	
29. Type and make of condenser equipment.....	
30. Rated capacity of condenser equipment.....	
31. Type of air pump, jacket pump, and reheater pump (direct or independently driven).....	
32. Type of governing apparatus.....	
33. Diameter of steam cylinders, inches.....	
34. Stroke of pistons, feet.....	
35. Clearance in terms of piston displacement, per cent:	
(a) Clearance, crank end.....	
(b) Clearance, head end.....	
36. Horsepower constant, steam cylinder:	
(a) Head end.....	
(b) Crank end.....	
37. Area of steam-cylinder jacketed surface, square feet.....	
38. Area of interior surface, square feet.....	
39. Steam-cylinder ratio.....	
46. Pressure above atmosphere in steam pipe near throttle, by gage...lb. per sq. in.	
(a) Corresponding absolute pressure.....	lb. per sq. in.
(b) Maximum pressure above atmosphere, steam-pipe diagram near throttle.....	lb. per sq. in.

- (c) Minimum pressure above atmosphere, steam-pipe diagram near throttle.....lb. per sq. in.
47. Pressure in steam receiver by gage, pound per square inch.....
48. Pressure in exhaust pipe near engine by mercury column, in in. of mercury...
(a) Corresponding absolute pressure.....lb. per sq. in.
49. Pressure in jackets and reheater.....lb. per sq. in.
60. Temperature of steam near throttle.....°F.
61. Temperature of saturated steam near throttle.....°F.
62. Temperature of steam leaving steam receiver, if superheated.....°F.
63. Temperature of steam in exhaust pipe as observed.....°F.
64. Temperature of saturated steam in exhaust pipe corresponding to pressure in exhaust pipe.....°F.
70. Superheat, at throttle.....°F.
71. Moisture in steam.....per cent
72. Total steam and water consumed by engine as measured.....lb.
73. Total steam less water consumed.....lb.
74. Correction factor conforming to conditions agreed upon.....lb.
75. Equivalent total steam consumed, conforming to conditions.....lb.
76. Steam and water (or superheated steam) consumed, per hour as measured...lb.
80. Steam less water (or superheated steam) consumed, per hour.....lb.
81. Equivalent steam conforming to conditions consumed, per hour.....lb.

Power Input

90. Steam cylinder—indicated horsepower developed, whole engine.....i.hp.
91. High-pressure steam cylinder, crank end.....i.hp.
head end.....i.hp.
92. Low-pressure steam cylinder, crank end.....i.hp.
head end.....i.hp.
104. Steam less water (or superheated steam) consumed per indicated horsepower (Item 80 ÷ Item 90).....lb.
105. Equivalent steam consumed per indicated horsepower (Item 81 ÷ Item 90).....lb.
106. Steam less water (or superheated steam) consumed per 100 cu. ft. of air or gas compressed at intake pressure and temperature ($100 \times \text{Item 73} \div \text{Item 67}$).....lb.
107. Equivalent steam consumed per 100 cu. ft. of air or gas compressed at intake pressure and temperature ($100 \times \text{Item 75} \div \text{Item 67}$).....lb.

NOTE.—Additional items applying only when the *driving element is an internal-combustion engine* are given in A.S.M.E. Power Test Code for Displacement Compressors and Blowers (Table 4), pp. 14 and 15.

CHAPTER XVIII

TESTING OF REFRIGERATION PLANTS

Refrigerating machines present a most interesting example of the conversion of heat energy. In the simplest forms these machines consist of a compressor driven by a steam engine, or other motive power, serving to compress a gas or vapor as the case may be. This gas or vapor is then passed under pressure through a surface condenser, where the cooling water absorbs the heat generated in the work of compression and then passes into an **expanding vessel** into which it discharges at a very low temperature. Now in order to vaporize any liquid, it is necessary to maintain a continual application of heat in order to bring about this physical change. To convert a unit weight of liquid to a unit weight of vapor at the same pressure, the heat required is always a constant quantity for the same liquid. Thus, as a familiar example, to convert a pound of water at "atmospheric" pressure and 212°F. into steam at the same pressure and temperature requires the application of 970 B.t.u.; and, conversely, to condense a pound of steam at this same pressure and temperature, it is necessary to abstract 970 B.t.u. by contact with a cold body. Steam as the working medium in a refrigerating machine would be commercially impracticable, because the lowest temperature resulting from **actual condensation** in a workable plant would be very much above the freezing point of water;¹ but there are a number of liquids which have a very much lower boiling point than water. Of these ammonia, NH_3 , carbon dioxide, CO_2 , and sulphur dioxide, SO_2 , are successfully used for purposes of refrigeration. The use of all these depends on the absorption of their latent heat in their conversion from a vapor or gas to the liquid condition. In practice the refrigerants most commonly used are anhydrous ammonia and sulphur dioxide, although carbon dioxide is also frequently employed. The latter is preferred usually where ammonia gas might be dangerous or otherwise objectionable.

In the simplest form of refrigerating plant the necessary machinery consists of (1) a **compressor** to raise the gas to the necessary pressure; (2) a **surface condenser** to absorb by means of cooling water the heat generated by the mechanical work of compression; and (3) an **expanding or evaporating vessel** where the liquid is reevaporated into a gas and, of

¹ A vacuum system of refrigeration using water as the cooling medium is explained in Moyer and Fittz, "Refrigeration," 2d ed., p. 32, McGraw-Hill Book Company, Inc., 1932.

course, absorbs heat in the operation.¹ A very simple refrigerating machine is shown in Fig. 291. It consists of the compressor *C* discharging gas under pressure through the pipe *P* into the condensing coil *D*, consisting in this simple apparatus of a coil of pipe in a tank through which the cooling water circulates. An expansion valve *V* serves for reducing the pressure and evaporating the liquid coming from *D*. The expanding vessel or **evaporator** *E* consists of a coil of pipe immersed in a tank containing the liquid to be cooled. Drops of liquid accumulate

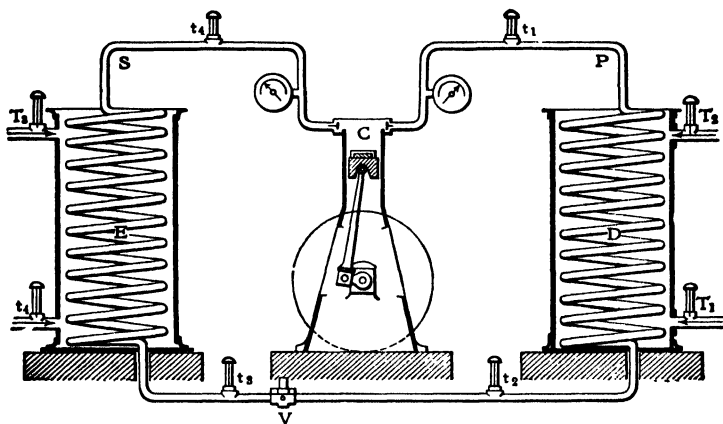


FIG. 291. . . Typical refrigerating apparatus.

in the bottom coils of the condenser *D*, to be discharged through the expansion valve *V* into the evaporator *E*. Since the compressor receives its supply of gas from the evaporator, the pressure in the latter must be less than in the condenser. On this account, then, the liquid after expanding will begin to boil and will absorb heat from the surrounding liquid in its transformation into a gas. In such a process the temperature of the cooling liquid may become very low. The refrigerating liquid in the evaporator will be entirely gasified or vaporized and returns finally to the compressor *C* in this state through the suction pipe *S*, thus completing the cycle of operations.

Anhydrous ammonia is most commonly used in large refrigerating plants. It is preferable to many other fluids because of its comparatively high latent heat and low pressure of vaporization.

A typical **commercial refrigerating plant** for making ice and operating with a horizontal ammonia compressor is shown in Fig. 292. The same descriptive letters used in Fig. 291 serve again for marking the important parts.

The efficiency of a refrigerating machine depends upon the difference between the extremes of temperature, but, unlike heat engines, it has the

¹ Modern refrigerating equipment is explained in detail in Moyer and Fittz, "Refrigeration," 2d ed., McGraw-Hill Book Company, Inc., New York, 1932.

greatest efficiency when the range of temperatures is small and when the final temperature is high.

When a change of volume of a saturated vapor is made under constant pressure in the presence of an excess of the liquid, the temperature remains constant. In this case the addition or absorption of heat to produce the change of volume causes an increase or decrease in the amount of the liquid mixed with the vapor. Vapors, even when saturated, if no longer in contact with their liquids, having heat added either

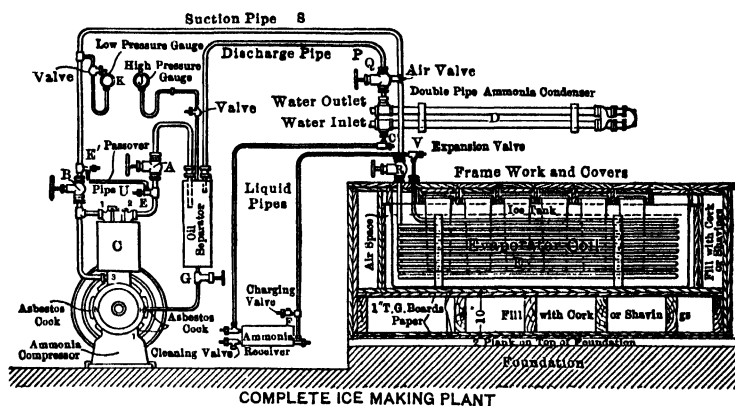


FIG. 292.—Refrigerating plant with ammonia compressor.

by compression (by mechanical force) or from an external source of heat, will behave practically like permanent gases and will become **superheated**. On this account refrigerating machines using liquefiable gas will give results differing according to the conditions of operation, depending primarily upon the state of the gas, that is, whether it remains constantly saturated or is superheated during a part of the cycle. Some ammonia plants are operated with an excess of liquid present during compression so that superheating is prevented. This is known in practice as the “wet” or “cold” system of compression.

TABLE XXVIII.—DENSITY AND SPECIFIC HEAT OF AMMONIA

At temperature, °F.....	15	20	30	40	50	60	70
Density, lb. per cu. ft.....	0.1524	0.1692	0.2073	0.2518	0.3036	0.3635	0.4325
Specific heat of liquid.....	1.078	1.072	1.060	1.048	1.036	1.024	1.012

Table XXVIII gives values of the density of saturated ammonia vapor and the specific heat of liquid ammonia.

Tables XXIX and XXX include the important data of the properties of saturated and superheated ammonia.

More complete tables of the properties of both saturated and superheated ammonia, prepared by the U. S. Bureau of Standards, are given in “Refrigeration” by Moyer and Fittz, 2d ed., pages 492 to 512.

TABLE XXIX.—PROPERTIES OF SATURATED AMMONIA
(Condensed from tables prepared by U. S. Bureau of Standards)

Pressure (abs.), lb. per sq. in.	Temper- ature, °F.	Specific volume, cu. ft. per lb.	Heat content or enthalpy, B.t.u. per lb.			Entropy		
			Liquid	Vapor (total)	Evapo- ration	Liquid	Evapo- ration	Vapor (total)
<i>p</i>	<i>t</i>	<i>v_g</i>	<i>h_f</i>	<i>h_g</i>	<i>h_{fg}</i>	<i>s_f</i>	<i>s_{fg}</i>	<i>s_g</i>
10.0	— 41.34	25.81	— 1.4	597.1	598.5	—0.0034	1.4310	1.4276
20.0	— 16.64	13.50	25.0	606.2	581.2	0.0578	1.3122	1.3700
30	— 0.57	9.236	42.3	611.6	569.3	0.0962	1.2402	1.3364
40	11.66	7.047	55.6	615.4	559.8	0.1246	1.1879	1.3125
50	21.67	5.710	66.5	618.2	551.7	0.1475	1.1464	1.2939
60	30.21	4.805	75.9	620.5	544.6	0.1668	1.1119	1.2787
70	37.70	4.151	84.2	622.4	538.2	0.1835	1.0823	1.2658
80	44.40	3.655	91.7	624.0	532.3	0.1982	1.0563	1.2545
90	50.47	3.266	98.4	625.3	526.9	0.2115	1.0330	1.2445
100	56.05	2.952	104.7	626.5	521.8	0.2237	1.0119	1.2356
110	61.21	2.693	110.5	627.5	517.0	0.2348	0.9927	1.2275
120	66.02	2.476	116.0	628.4	512.4	0.2452	0.9749	1.2201
130	70.53	2.291	121.1	629.2	508.1	0.2548	0.9584	1.2132
140	74.79	2.132	126.0	629.9	503.9	0.2638	0.9430	1.2068
150	78.81	1.994	130.6	630.5	499.9	0.2724	0.9285	1.2009
160	82.64	1.872	135.0	631.1	496.1	0.2804	0.9148	1.1952
170	86.29	1.764	139.3	631.6	492.3	0.2881	0.9019	1.1900
180	89.78	1.667	143.3	632.0	488.7	0.2954	0.8896	1.1850
190	93.13	1.581	147.2	632.4	485.2	0.3024	0.8778	1.1802
200	96.34	1.502	150.9	632.7	481.8	0.3090	0.8666	1.1756
225	103.87	1.336	159.7	633.3	473.6	0.3246	0.8405	1.1651
250	110.80	1.202	168.0	633.8	465.8	0.3388	0.8167	1.1555
275	117.22	1.091	175.6	634.0	458.4	0.3519	0.7947	1.1466
300	123.21	0.999	182.9	634.0	451.1	0.3642	0.7741	1.1383

Leakages of ammonia gas are very objectionable and may be dangerous. A small leak may be found by burning a little sulphur at the end of a stick of wood about 15 inches long. A white vapor will be observed at the leak.

A.S.M.E. TEST CODE (1926) FOR REFRIGERATING SYSTEMS (ABRIDGED)

The Test Code for Refrigerating Systems is intended for use in the determination of the performance of compression systems in which compressors of the reciprocating type are used as well as absorption machines. In so far as the fundamental operations of all the systems are in common, general rules will be laid down, and where the systems differ separate rules will be given. For the test of the driving element in compression systems, the A.S.M.E. Power Test Codes for Steam Engines, Steam Turbines, Internal-combustion Engines, etc., should be followed.

TABLE XXX.—PROPERTIES OF SUPERHEATED AMMONIA
(Condensed from tables prepared by U. S. Bureau of Standards)

v = specific volume, in cubic feet per pound; h_g = heat content or enthalpy, in B.t.u. per pound; S = total entropy.

Pressure (abs.), lb. per sq. in. (p)	Temper- ature of satu- rated vapor, °F. (t)	Superheat, degrees Fahrenheit								
		20	40	60	80	100	150	200	250	300
20	− 16.6 v	14.78	15.45	16.12	16.78	17.43	19.05	20.66		
	h_g	626.4	637.0	647.5	658.0	668.5	694.7	721.2		
	S	1.4138	1.4356	1.4562	1.4760	1.4950	1.5399	1.5817		
30	− 0.6 v	9.731	10.20	10.65	11.10	11.55	12.65	13.73	14.81	
	h_g	623.5	634.6	645.5	656.2	666.9	693.5	720.3	747.5	
	S	1.3618	1.3845	1.4059	1.4261	1.4456	1.4911	1.5334	1.5727	
40	11.7 v	7.203	7.568	7.922	8.268	8.609	9.444	10.27	11.08	
	h_g	620.4	632.1	643.4	654.4	665.3	692.3	719.4	746.8	
	S	1.3231	1.3470	1.3692	1.3900	1.4098	1.4561	1.4987	1.5382	
50	21.7 v		5.988	6.280	6.564	6.843	7.521	8.185	8.840	9.489
	h_g		629.5	641.2	652.6	663.7	691.1	718.5	746.1	774.0
	S		1.3169	1.3399	1.3613	1.3816	1.4286	1.4716	1.5118	1.5500
60	30.2 v		4.933	5.184	5.428	5.665	6.239	6.798	7.348	7.892
	h_g		626.8	639.0	650.7	662.1	689.9	717.5	745.3	773.3
	S		1.2913	1.3152	1.3373	1.3581	1.4058	1.4493	1.4897	1.5281
70	37.7 v		4.177	4.401	4.615	4.822	5.323	5.807	6.282	6.750
	h_g		623.9	636.6	648.7	660.4	688.7	716.6	744.5	772.7
	S		1.2688	1.2937	1.3166	1.3378	1.3863	1.4302	1.4710	1.5095
80	44.4 v			3.812	4.005	4.190	4.635	5.063	5.481	5.894
	h_g			634.3	646.7	658.7	687.5	715.6	743.7	772.1
	S			1.2745	1.2981	1.3199	1.3692	1.4136	1.4547	1.4933
90	50.5 v			3.353	3.529	3.698	4.100	4.484	4.859	5.288
	h_g			631.8	644.7	657.0	686.3	714.7	743.0	771.5
	S			1.2571	1.2814	1.3038	1.3539	1.3988	1.4401	1.4789
100	56.1 v			2.985	3.149	3.304	3.672	4.021	4.361	4.695
	h_g			629.3	642.6	655.2	685.0	713.7	742.2	770.8
	S			1.2409	1.2661	1.2891	1.3401	1.3854	1.4271	1.4660
110	61.2 v				2.837	2.981	3.321	3.642	3.954	4.259
	h_g				640.5	653.4	683.7	712.8	741.5	770.2
	S				1.2519	1.2755	1.3274	1.3732	1.4151	1.4543

TABLE XXX.—PROPERTIES OF SUPERHEATED AMMONIA (*Continued*)

Pressure (abs.), lb. per sq. in. (p)	Temper- ature of satu- rated vapor, °F. (t)	Superheat, degrees Fahrenheit								
		20	40	60	80	100	150	200	250	300
120	66.0 <i>v</i>				2.576	2.712	3.029	3.326	3.614	3.895
	<i>h_g</i>				638.3	651.6	682.5	711.8	740.7	769.6
	<i>S</i>				1.2386	1.2628	1.3157	1.3620	1.4042	1.4435
130	70.5 <i>v</i>				2.355	2.484	2.781	3.059	3.326	3.587
	<i>h_g</i>				636.0	649.7	681.2	710.9	739.9	769.0
	<i>S</i>				1.2260	1.2509	1.3048	1.3516	1.3941	1.4336
140	74.8 <i>v</i>				2.166	2.288	2.569	2.830	3.080	3.323
	<i>h_g</i>				633.8	647.8	679.9	709.9	739.2	768.3
	<i>S</i>				1.2140	1.2396	1.2945	1.3418	1.3846	1.4243
150	78.8 <i>v</i>					2.118	2.385	2.631	2.866	3.095
	<i>h_g</i>					645.9	678.6	708.9	738.4	767.7
	<i>S</i>					1.2289	1.2849	1.3327	1.3758	1.4157
160	82.6 <i>v</i>					1.969	2.224	2.457	2.679	2.895
	<i>h_g</i>					643.9	677.2	707.9	737.6	767.1
	<i>S</i>					1.2186	1.2757	1.3240	1.3675	1.4076
180	89.8 <i>v</i>					1.720	1.955	2.167	2.367	2.561
	<i>h_g</i>					639.9	674.5	705.9	736.1	765.8
	<i>S</i>					1.1992	1.2586	1.3081	1.3521	1.3926
200	96.3 <i>v</i>					1.520	1.740	1.935	2.118	2.295
	<i>h_g</i>					635.6	671.8	703.9	734.5	764.5
	<i>S</i>					1.1809	1.2429	1.2935	1.3382	1.3791
220	102.4 <i>v</i>						1.564	1.745	1.914	2.076
	<i>h_g</i>						669.0	701.9	732.9	763.2
	<i>S</i>						1.2281	1.2801	1.3255	1.3668
240	108.1 <i>v</i>						1.416	1.587	1.745	1.895
	<i>h_g</i>						666.1	699.8	731.3	762.0
	<i>S</i>						1.2145	1.2677	1.3137	1.3554

Measurements.—The principal measurements and quantities determined in a test are:

(a) The quantity of the refrigerant circulated; (b) the various temperatures of the refrigerant in the cycle; (c) the various pressures of the refrigerant in the cycle; (d) the energy required to operate the driving element and the various auxiliary appa-

tus; (e) the various temperatures of condensing and cooling water; (f) the quantity of condensing and cooling water; (g) quantity, quality, and pressure of steam used in the steam cylinder or generator; (h) quantity of oil passing through the compressor; (i) quantity, composition of, and density of brine used; (j) temperature of brine entering and leaving the evaporator; (k) quality of liquor in absorption plants.

Instruments and apparatus necessary in carrying out the test are:

(a) Suitable meter for measuring liquid refrigerant on the high- or intermediate-pressure side where same passes to the expansion coils of the brine cooler or elsewhere; (b) platform scales and suitable tanks for measuring water and condensed steam; (c) water meters, calibrated tanks, or tank and platform scale for measuring condensing and cooling water; (d) pressure gages and vacuum manometers including a suitable U-tube or mercury column to be connected at the low-pressure suction of the compressor. It is recommended that this be used in all cases rather than a pressure or vacuum gage for suction pressures under 20 pounds' gage; (e) calibrated thermometers and thermometer mercury wells of sufficient lengths to extend beyond the centers of the pipes. Thermometers used for taking temperatures of brine shall be graduated in $^{\circ}\text{F}$. and readings estimated for $^{\circ}\text{F}$.; (f) barometer and psychrometer when necessary; (g) revolution counter or other accurate speed-measuring device; (h) graduated stroke indicator for direct-acting pumps; (i) calibrated hydrometer for specific gravity of brine; (j) calorimeter for determining the quality of the steam; (k) steam and ammonia indicators; (l) planimeter; (m) dead-weight gage tester; (n) appropriate instruments for measuring current consumption (power) of electric motor-driven machines.

The *thermometer wells* should be of steel for ammonia lines and of bronze for brine lines and should extend into the pipe a sufficient distance so that the bulb of the thermometer will extend at least to the center of the pipe. These wells should be filled with mercury to the point indicated on the stem of the thermometer. Insulation on the cold lines should extend even to the top of the well, and the thermometers pass through corks fitted to the wells and bored to accommodate the thermometers. The thermometer well must be placed at least 10 feet away from brine coolers or pumps so that the solution has uniform temperature where the readings are taken. Electric-resistance thermometers or thermocouples may be used in taking temperatures of brine as a check on mercury thermometers. In testing a plant where the load is light, a brine heater or a heater using waste condensing water flowing over the evaporating coils should be used.

For ammonia a special steel indicator must be provided which has steel indicator cocks with short separate connections to the ends. The indicator should be provided with a stop so that a light spring may be used for studying suction conditions.

In the compression system it is required under the Code that nothing but liquid shall enter the expansion valve and nothing but vapor shall enter the refrigerating machine.

In ammonia absorption systems the anhydrous ammonia should not contain over 3 per cent moisture. Samples should be drawn before and immediately after the test and determinations of moisture made.

Starting and Stopping.—The plant should be operated a sufficient length of time prior to the starting of the test to insure uniformity of conditions and, when these conditions are obtained, the test should be started and continued. It is essential to the accuracy of the test that all parts of the plant contain the same amount of heat in the refrigerant (estimated above some datum) at the end as at the beginning of the test. To accomplish this, all vessels containing liquid refrigerant should be supplied with suitable gage glasses so that the same quantity of fluid may be distributed alike at the beginning and end of the test and the quantities and temperature of the fluids in the various parts maintained as uniform as possible.

Duration of Tests.—Each run shall be continued over a period of time not less than t hours, t to be calculated from the following formula:

$$tw_a = 3w_b,$$

where w_a = approximate weight of anhydrous refrigerant circulated, in pounds per hour.

w_b = total weight of anhydrous refrigerant contained in the system, in pounds.

Fifteen-minute readings will be sufficient except where there is considerable fluctuation in the readings, in which case more frequent readings must be taken to insure good averages.

Each indicator card should be marked with the date, time of day, cylinder and end of cylinder, identification mark, and strength of indicator spring, as pointed out in the Test Code for Reciprocating Steam Engines (page 355). To assist in the uniformity of operation a chart should be plotted during the test of the principal quantities, including the estimated tonnage.

The data and results should be tabulated in accordance with the form shown in the following tables, adding items not provided for and omitting items not needed to conform to the object in view. Unless otherwise indicated, the items refer to the numerical readings which are recorded in the log.

Calculation of Results.—The now generally adopted recommendations of the Joint Committee of the A.S.M.E. and the American Society of Refrigerating Engineers on **Standard Tonnage Basis for Refrigeration** are as follows:

- a. A standard ton of refrigeration is 288,000 B.t.u.
- b. The standard commercial ton of refrigeration is at the rate of 200 B.t.u. per minute or 12,000 B.t.u. per hour.
- c. The standard rating of a refrigerating machine¹ using liquefiable vapor is the number of standard commercial tons of refrigeration it performs under adopted refrigerant pressures.

¹ A refrigerating machine is the compressor cylinder of the compression refrigerating system or the absorber, liquor pump, and generator of the absorption refrigerating system.

There shall be 9°F. (5°C.) subcooling of the liquid refrigerant entering the expansion valve and 9°F. (5°C.) superheating of the vapor entering the refrigerating machine, the points at which the subcooling and the superheating are determined to be within 10 feet of the expansion valve and refrigerating machine, respectively.

The inlet pressure is that which corresponds to a saturation temperature of 5°F. (−15°C.).

The outlet pressure is that which corresponds to a saturation temperature of 86°F. (30°C.).

These pressures are measured outside and within 10 feet of the refrigerating machine, distances which are measured along the inlet and outlet pipes, respectively.

The **capacity** of the system shall be expressed in tons of refrigeration per 24 hours as calculated from the weight of liquid refrigerant and the amount of available cooling effect produced in the evaporator. This method shall be employed in the case of all refrigerants where the physical properties have been determined and where tables of such properties are recognized as sufficiently accurate. An ammonia chart is given in the Appendix.¹

The weight of refrigerant circulated shall be determined by a suitable liquid meter. As an alternate method the weight of liquid refrigerant shall be calculated from the volume as obtained from a pair of calibrated receivers or by direct weighing.

The amount of **available refrigerating effect** shall be the weight of refrigerant circulated per hour multiplied by the difference in heat content on entering and leaving the evaporator. Where there is more than one evaporator in which different pressures exist, the weight of refrigerant to each evaporator and the total heat from the vapor must be determined separately and the total refrigerating effect may then be computed by the following formula:

$$Q_e = W_a (h_{ge} - h_{gz}),$$

where Q_e = total refrigerating effect per hour.

W_a = weight of anhydrous refrigerant circulated, in pounds per hour.

h_{ge} = total heat in the vapor leaving the evaporator.

h_{gz} = total heat in the liquid at the expansion valve.

The **refrigeration effect (R.E.)** expressed in standard commercial tons of refrigeration is

$$R.E. = \frac{Q_e}{12,000}.$$

¹ Properties of a large number of refrigerants are given in Moyer and Fittz, "Refrigeration," 2d ed., pp. 73 to 105 and 492 to 512.

Theoretical Horsepower Required to Compress Adiabatically.—The work done in the cylinder of the compressor per pound of refrigerant circulated expressed in thermal units may be found by referring to either the Mollier chart or the tables adopted and published by the American Society of Refrigerating Engineers.

The theoretical horsepower hp_t is

$$hp_t = \frac{(h_{g1} - h_{g2}) \times W_a \times 778}{33,000 \times 60},$$

where h_{g1} = heat content of vapor after adiabatic compression.

h_{g2} = heat content of vapor at compressor inlet.

W_a = weight of refrigerant circulated, in pounds per hour.

Indicated Horsepower.—The indicated horsepower for each end of the cylinder is found by using the formula:

$$i.hp. = \frac{p_m LAN}{33,000},$$

where p_m = mean effective pressure, in pounds per square inch.

L = length of the stroke, in feet.

A = area, in square inches, of the piston less the area of the piston rod, if any.

N = number of revolutions per minute.

The total horsepower of the cylinder is the sum of the horsepower developed in the two ends.

Mechanical Efficiency.—Mechanical efficiency is the ratio of the indicated horsepower in the compressor cylinders to the indicated horsepower in the power cylinders in the case of a steam-driven or internal-combustion-driven compressor, and to the brake horsepower delivered to the shaft in the case of a motor-driven machine.

Compression Efficiency.—Compression efficiency of a *single-cylinder* compressor is the ratio of horsepower required to compress adiabatically all the vapor delivered by the compressor to the horsepower developed within the compressor cylinder as shown by the indicator cards.

Mean Effective Pressure.—The mean effective pressure should be found by dividing the area of the indicator diagram in square inches as determined with a planimeter by the length of the diagram in inches and multiplying the quotient by the scale of the indicator spring. If a planimeter is not available, the approximate mean effective pressure may be determined by finding the average height of the diagram in inches as obtained by averaging a suitable number of equally spaced ordinates, at least 20, measured between the lines of the forward and return strokes, and then multiplying this average by the calibrated scale of the spring. The length of the indicator diagram is the measured distance along the atmospheric-pressure line between ordinates erected perpendicular to it and passing through the ends of the indicator diagram.

In the case of fluids not having published charts or tables, the following method may be used to calculate the theoretical horsepower required to compress the vapor adiabatically at the average rate for a single stage or for any stage of multiple-stage compressors:

$$\text{hp.} = \frac{144k}{33,000(k-1)} p_1 v_1 \left[\frac{p_2^{\frac{k-1}{k}}}{p_1} - 1 \right]$$

where v_1 = volume compressed, in cubic feet per minute.

p_1 = absolute intake pressure, in pounds per square inch.

p_2 = absolute discharge pressure, in pounds per square inch.

k = the exponent in $pv^k = \text{constant}$.

Gross Horsepower.—The gross horsepower is the indicated horsepower in the steam or power cylinders in the case of a steam or internal-combustion engine-driven compressor; the electrical horsepower multiplied by the motor efficiency in the case of a direct-connected motor-driven compressor; and the brake horsepower delivered to the compressor shaft in the case of a belt-driven or gear-driven compressor. If motor efficiency determined under the A.I.E.E. rules is used, the value obtained will be a conventional one and will always be higher than the actual efficiency of the motor. An agreement must therefore be made by those interested in the test as to the tolerance to be applied to the "conventional efficiency" to convert the figure into actual efficiency.

Electrical Horsepower.—The electrical horsepower of a motor is found by dividing the input at the terminals expressed in kilowatts by the constant 0.746. In the case of an alternating-current motor, the input determined, whether expressed in electrical horsepower or kilowatts, should be the total input. When the power for excitation is taken directly from the compressor shaft, the total input is that indicated at the alternating-current motor terminals. When the power for excitation is furnished from an independent source, the total electrical horsepower used in calculating the gross horsepower (page 478) is that indicated at the motor terminals plus an equivalent power required to run the exciter.

Volumetric Efficiency.—The volumetric efficiency is the ratio of the capacity of the compressor to displacement. The capacity is the actual amount of vapor compressed and delivered, expressed in cubic feet per minute at intake temperature and pressure.

The **compression efficiency** for any particular cylinder of a multistage compressor is the ratio of the horsepower required to compress adiabatically all of the vapor delivered by the compressor through the observed pressure range for the particular cylinder in question to the horsepower developed in the particular cylinder in question as shown by the indicator cards. The two factors involved in this ratio are defined on page 470.

Overall Pump Efficiency.—The method of computing the overall efficiency is described in the Test Code for Displacement Compressors and Blowers (page 449).

Condenser Performance.—The condenser is required to remove the total heat of the vapor drawn into the compressor plus the heat of compression. It is therefore incorrect to consider the removal of 12,000 B.t.u. per hour by the condensing water as equivalent to the standard commercial ton as produced in the evaporator. The total heat removed by the condensing water shall be calculated from the observed data when the enclosed type of condensers, either shell and tube or double pipe, is used. The total heat may also be calculated from the quantity of refrigerant circulated. In this case

$$Q_e = W_a(h_{g3} - h_{g4}),$$

where Q_e = heat removed per hour.

h_{g3} = heat content of vapor entering condenser.

h_{g4} = heat content of liquid leaving condenser.

W_a = weight of refrigerant condensed per hour.

Determination of Heat Transfer in Condensers.—The work done in the condenser may be divided into three stages: (1) the removal of superheat; (2) the liquefaction of the refrigerant; and (3) the subcooling of the condensed liquid. It is therefore to be expected that the rate of heat transfer is not the same in all parts of the condenser. The *logarithmic mean difference* (page 492) is based upon a constant temperature of liquefaction throughout the condenser on the assumption that the heat transfer is uniform throughout the condenser. While the calculation of heat transfer per square foot of refrigerant surface per hour per degree of logarithmic mean difference does not give entirely satisfactory results, the determination of heat transfer shall be calculated in this manner.

$$\text{Logarithmic mean difference} = \frac{t_o - t_i}{\log_e \frac{t_e - t_i}{t_e - t_o}},$$

where t_e = temperature of liquefaction corresponding to condenser pressure.

t_i = temperature of incoming water.

t_o = temperature of outgoing water.

Evaporator Performance.—The evaporator performance shall be calculated from the observed data. In this case

$$Q_e = W_a(h_{g5} - h_{g6}),$$

in which Q_e = heat removed per hour.

h_{g5} = heat content of vapor leaving evaporator.

h_{g6} = heat content of liquid entering expansion valve.

W_a = weight of refrigerant entering evaporator, in pounds per hour.

Determination of Heat Transfer in Evaporator.—Brine coolers. Brine-cooler performance shall be expressed in terms of heat transfer per square foot of cooling surface per hour per degree of logarithmic mean difference.

$$\text{Logarithmic mean difference} = \frac{t_i - t_o}{\log_e \frac{t_i - t_v}{t_o - t_v}},$$

where t_o = temperature of brine out of evaporator.

t_i = temperature of brine to evaporator.

t_v = temperature of vaporization corresponding to evaporator pressure.

The performance of ice tanks and all other forms of expansion coils or apparatus will not be specifically covered in this Code, which is intended to include the main items of the refrigerating cycle rather than the products of the cycle.

Water Horsepower.—The water-horsepower output at observed total suction and discharge pressure is

$$\text{w.hp.} = \frac{(\text{pounds of liquid per minute}) \times (\text{total head in feet})}{33,000}.$$

The gross or brake horsepower input shall be observed for all pump-driving elements and the efficiencies calculated.

Heat Balance.—A heat balance shall be made of the refrigerating cycle, of which the following is an illustration of all the items required in a "compound-cycle" compression system. As a heat balance is impossible unless extreme care has been taken in making the measurements, it will be found necessary to have most accurate thermometers for observing the temperature of water on and off condensers as well as for taking temperatures of brine. It is also necessary to have indicator cards taken at uniform intervals, at least every 30 minutes, in order to determine the heat equivalent of work done in the compressor.

The general formulas for heat balance are as follows:

Compression cycle: $Q_c + Q_w = Q_1 + Q_3$

Absorption cycle: $Q_r + Q_s = Q_1 + Q_2 + Q_3$

where Q_c = heat absorbed by evaporating refrigerant.

Q_w = heat equivalent of work in compressor.

Q_s = heat imparted by steam in generator.

Q_1 = heat rejected in condenser.

Q_2 = heat rejected in absorber.

Q_3 = heat rejected or radiated in addition to Q_1 and Q_2 .

For purposes of illustration the following list of quantities involved in the computation of the heat balances of *compound compression* systems is given.

Heat Absorbed	Heat Rejected
(In B.t.u. per Hour)	
a. Heat absorbed in evaporator.	n. Heat rejected by hot surfaces of low-pressure compressor.
b. Heat entering evaporator insulation.	o. Heat rejected from low-pressure discharge main between low-pressure compressor and intermediate vapor cooler.
c. Heat absorbed in low-pressure suction main.	p. Heat rejected in intermediate vapor cooler.
d. Heat absorbed in low-pressure suction trap.	q. Heat rejected to engine room by intermediate vapor cooler.
e. Heat equivalent of work done in compressor.	r. Heat rejected in discharge main from intermediate vapor cooler to intermediate liquid receiver.
f. Heat absorbed from engine room through cold surface of low-pressure compressor.	s. Heat rejected by hot surfaces of high-pressure compressor.
g. Heat absorbed through surface of intermediate liquid receiver.	t. Heat rejected by high-pressure discharge main and oil separator between machine and condensers.
h. Heat absorbed through surface of intermediate-pressure liquid line.	u. Heat rejected in ammonia condensers.
i. Heat absorbed from engine room through surface of high-pressure suction main.	v. Heat rejected in liquid cooler.
j. Heat absorbed through cold surfaces of high-pressure compressor.	w. Same as k.
k. Heat absorbed or rejected through condenser shells.	x. Same as l.
l. Heat absorbed or rejected through receivers.	y. Same as m.
m. Heat absorbed or rejected in high-pressure liquid line.	

TABLE XXXI.—DATA AND RESULTS OF TEST ON POWER-DRIVEN REFRIGERATING UNIT

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

Description, Dimensions of Compressor, Condenser, Etc.

7. Type of compressor (single or multiple stage, and kind of refrigerant).....
8. Type of compressor valves.....
9. Method of driving compressor.....
10. Method of volume control.....
11. Rated discharge (gage) pressure.....lb. per sq. in.
12. Rated speed.....r.p.m.
13. Rated displacement.....cu. ft. per min.
14. Rated output expressed in standard tons of refrigeration.....
15. Type of intercoolers.....

	First Inter- cooler	Second Inter- cooler
16. Area of water-cooled surface.....sq. ft.		
	First Stage	Second Stage
17. Diameter of compressor cylinders.....in.		
18. Stroke of pistons.....ft.		
19. Diameter of piston rods or tail rods.....in.		
20. Clearance in terms of piston displacement in per cent		
(a) Clearance, head end.....		
(b) Clearance, crank end.....		
(c) Clearance, average.....		
21. Cylinder ratio based on piston displacement		
(a) First stage to second stage.....		
22. Horsepower constant, cylinder:		
(a) Head end (stroke $\times$ net piston area $\div$ 33,000)....		
(b) Crank end (stroke $\times$ net piston area $\div$ 33,000)....		
23. Area of compressor cylinder jacketed surface.....sq. ft.		
24. Diameter of intake pipe.....in.		
25. Diameter of final discharge pipe.....in.		

NOTE.—For description, dimensions, etc., of compressor-driving element, see pp. 478 and 479.

26. Type of condenser.....	
27. Area of condenser surface on refrigerant side.....sq. ft.	
28. Type of evaporator.....	
29. Area of evaporator surface on refrigerant side.....sq. ft.	
30. Type of water pumps.....	
31. Size of water pumps, No. 1, No. 2.....gal. per min.	
32. Type of brine pumps.....	
33. Size of brine pumps, Nos. 1, 2, 3.....gal. per min.	
34. Type of cooling tower.....	
35. Rated capacity of cooling tower.....	
36. Type of liquid after-cooler.....	
37. Cooling surface of liquid after-cooler.....sq. ft.	
38. Type of intermediate liquid cooler.....	
39. Size of intermediate liquid cooler..... diam..... height	

General Data

40. Composition of brine.....	
41. Specific gravity of brine used compared with water at 39.1°F.*	
42. Specific heat of brine used.....	
43. Specific gravity of liquid refrigerant as measured.....	
44. Kind of cooling water used (well, salt, river).....	
45. Specific gravity of cooling water.....	
46. Specific heat of cooling water.....	

Test Data and Results

61. Duration of test.....	hr.
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* Temperature of the maximum density of water.

Average Pressures

62. Barometric pressure.....in. of mercury at 32°F.
 (a) Corresponding absolute pressure.....in. of mercury at 32°F.
63. Pressure by mercury manometer at intake near cylinder first stage.in. of mercury at 32°F.
 (a) Corresponding absolute pressure.....lb. per sq. in. at 32°F.

	First Stage	Second Stage
64. Pressure by gage in discharge pipe near cylinder, lb. per sq. in.....		
(a) Corresponding absolute pressure, lb. per sq. in....		
65. Pressure in intercooler by gage, lb. per sq. in.....		
(a) Corresponding absolute pressure, lb. per sq. in....		
66. Condenser pressure by gage, lb. per sq. in.....		
(a) Corresponding absolute pressure, lb. per sq. in....		
67. Pressure in evaporator, by mercury manometer, in. of mercury at 32°F.....		
(a) Corresponding absolute pressure, lb. per sq. in....		
68. Pressure at inlet of cooling-water pumps by gage, lb. per sq. in.....		
69. Pressure at outlet of cooling-water pumps by gage, lb. per sq. in.....		
70. Pressure at inlet of brine pump by gage, lb. per sq. in.		
71. Pressure at outlet of brine pump by gage, lb. per sq. in.		

Average Temperatures.....

76. Engine-room temperature.....°F.
77. Condenser-room temperature.....°F.
78. Temperature of outside air.....°F.
79. Temperature of wet bulb (p. 441), outside air.....°F.
80. Temperature of vapor near intake (to be measured not more than 2 ft. from the compressor inlet and should be between 6 and 12°F. of superheat), °F.
- 80a. Temperature of vapor from evaporator (to be measured not more than 2 ft. from outlet), °F.
81. Temperature of vapor near discharge port, °F.
82. Temperature of vapor at inlet to intermediate vapor cooler, °F.
83. Temperature of vapor at outlet to intermediate vapor cooler, °F.
84. Temperature of liquid from condensers.....°F.
85. Temperature of liquid from liquid receivers.....°F.
86. Temperature of liquid from liquid cooler.....°F.
87. Temperature of liquid from intermediate liquid cooler.....°F.
88. Temperature of liquid at measuring point.....°F.
89. Temperature of jacket cooling water, inlet.....°F.
90. Temperature of jacket cooling water, outlet.....°F.
91. Temperature of cooling water, intermediate vapor cooler inlet.....°F.
92. Temperature of cooling water, intermediate vapor cooler outlet.....°F.
93. Temperature of cooling water, liquid-cooler inlet.....°F.
94. Temperature of cooling water, liquid-cooler outlet.....°F.
95. Temperature of cooling water, condenser inlet.....°F.

96. Temperature of cooling water, condenser outlet.....	°F.
97. Temperature of water to cooling tower.....	°F.
98. Temperature of water from cooling tower.....	°F.
99. Temperature of brine, brine-cooler inlet.....	°F.
100. Temperature of brine, brine-cooler outlet.....	°F.
101. Temperature range through brine coolers by electric-resistance thermometers or by thermocouples.....	°F.

Total Quantities

107. Liquid delivered to evaporator.....	lb.
108. Brine circulated (at known quantities).....	gal.

Unit Quantities

115. Liquid delivered to evaporator (Item 107 ÷ Item 61).....	lb. per hr.
116. Brine circulated (Item 108 ÷ Item 61 × 60).....	gal. per min.
117. Cooling water supplied to condensers.....	gal. per min.

		First Stage	Second Stage
118. Cooling water supplied to jackets.....	gal. per min.		
119. Cooling water supplied to intermediate vapor cooler.....	gal. per min.		
120. Cooling water supplied to liquid cooler.....	gal. per min.		

Refrigerating Output

121. Available cooling effect (Q_e) (p. 469)	
122. (a) Standard commercial tons of refrigeration produced (Item 121 ÷ 12,000).....	tons
(b) tons of refrigeration, brine method as a check (Item 116 × Item 101 × Item 41 × 8.33 × Item 42 ÷ 200).....	tons

Speed

126. Total number of revolutions as shown by compressor counter.....	
127. Revolutions per minute Item 126 ÷ (Item 61 × 60).....	r.p.m.
128. Average piston speed.....	ft. per min.

Power of Compressor

129. Indicated horsepower of compressor cylinders, whole compressor.....	i.hp.
130. Low-pressure cylinder or cylinder if single-stage:	
(a) Crank end.....	i.hp.
(b) Head end.....	i.hp.
131. High-pressure cylinder:	
(a) Crank end.....	i.hp.
(b) Head end.....	i.hp.
132. Friction horsepower [Item 144 (Table XXXII), or Item 133 (Table XXXIII) — Item 129].....	i.hp.

Power Input

133 to 145. See pp. 479 and 480.

Economy Results

146. Gross or brake horsepower of compressor per ton of refrigeration.....	hp.
147. Indicated horsepower of compressor per ton of refrigeration [Item 129 ÷ Item 122(a)].....	hp.

Efficiency Results

148. Theoretical horsepower to compress adiabatically by calculation from tables or Mollier charts:

	First Stage	Second Stage
(a) Total heat contained in 1 lb. vapor at suction, B.t.u.		
(b) Total heat contained in 1 lb. liquid entering expansion valve, B.t.u.		
(c) Available heat of evaporation per lb. of refrigerant [Item 148(a) — Item 148(b)].....		
(d) Work required to compress 1 lb. refrigerant adiabatically. See Mollier chart for ammonia (Appendix), B.t.u.		
(e) Theoretical temperature of discharge vapor (see Mollier chart), °F.		
(f) Horsepower per ton for adiabatic compression, i.hp.		
149. Theoretical hp. to compress adiabatically by formula (page 470).....		hp.
150. Volumetric efficiency.....		per cent
151. Compression efficiency.....		per cent
152. Mechanical efficiency of compressor.....		per cent

Condenser

153. Heat removed per minute by condensing water (Item 96 — Item 95) × Item 117 × 8.33.....	B.t.u.
154. Heat removed as calculated from refrigerant condensed as a check.....	
155. Heat removed per hour (Item 153 × 60).....	B.t.u.
156. Logarithmic mean temperature difference, °F (p. 472).....	
157. Coefficient of heat transfer, $K = \text{Item 155} \div (\text{Item 156} \times \text{Item 27})$	

Evaporator

158. Heat removed per minute from brine (Item 122b × 200).....	B.t.u.
159. Heat removed per hour (Item 158 × 60).....	B.t.u.
160. Logarithmic mean temperature difference = °F.....	
161. Coefficient of heat transfer, $K = \text{Item 159} \div (\text{Item 160} \times \text{Item 29})$	

Auxiliary pumps:

(a) <i>Brine pumps</i>	
162. Gross or brake horsepower.....	b.hp.
163. Water horsepower.....	hp.
164. Overall pump efficiency.....	per cent
(b) <i>Water pumps</i>	
165. Water horsepower.....	hp.
166. Overall pump efficiency.....	per cent

TABLE XXXII.—ADDITIONAL ITEMS APPLYING ONLY WHEN THE DRIVING ELEMENT IS AN ELECTRIC MOTOR

Description, Dimensions, Etc., of Compressor Drive

47. Type of motor.....	
48. Rated power of motor.....	hp.
49. Volts.....	
50. Amperes.....	

- 51. Phase.....
- 52. Cycles.....per sec.
- 53. Revolutions per minute.....
- 54. Type and rating of exciter.....
 - (a) Voltage.....volts
 - (b) Current.....amp.

Power Input

- 133. Electric power to motor terminals.....kw.
- 134. { (a) Voltage per phase.....volts
 - (b) Current per phase.....amp.
- 135. Power factor (from meter).....
- 136. Exciter:
 - (a) Voltage.....volts
 - (b) Current.....amp.
- 137. Efficiency of exciter (from builder's test).....per cent
- 138. Electric horsepower to drive exciter.....hp.
- 139. Electric power input to armature of driving motor (Item 133 $\div$ 0.746).....hp.
- 140. Electric power input to armature of driving motor (Item 133 or calculated from Items 134 and 135).....kw.
- 141. Electric power input to field of driving motor (calculated from Item 136).....kw.
- 142. Total electric input to driving motor (Item 140 + Item 141).....kw.
- 143. Efficiency of driving motor (from builder's test).....per cent
- 144. Electric horsepower to drive compressor.....hp.

Economy Results

- 145. Electric horsepower of compressor per standard ton of refrigeration.....hp.

TABLE XXXIII.—ADDITIONAL ITEMS APPLYING ONLY WHEN THE DRIVING ELEMENT IS A STEAM ENGINE

Description, Dimensions, Etc., of Compressor Drive

- 47. Type of engine (simple or multiple expansion or uniflow).....
- 48. Type of steam valves.....
- 49. Auxiliaries (steam or electric drive).....
- 50. Type and make of condenser equipment.....
- 51. Rated capacity of condenser equipment.....
- 52. Type of air pump, jacket pump, and reheater pump (direct or independently driven).....
- 53. Type of governing apparatus.....
- 54. Diameter of steam cylinders.....in.
- 55. Stroke of pistons.....ft.
- 56. Clearance in terms of piston displacement
 - (a) Clearance, crank end.....per cent
 - (b) Clearance, head end.....per cent
- 57. Horsepower constant (p. 152), steam cylinder:
 - (a) Head end.....
 - (b) Crank end.....
- 58. Area of steam-cylinder jacketed surface.....sq. ft.
- 59. Area of interior surface.....sq. ft.
- 60. Steam-cylinder ratio (overall)

136. Steam less water (or superheated steam) consumed per indicated horsepower-hour of compressor (Item 124 ÷ Item 129).....	lb.
137. Equivalent steam consumed per indicated horsepower-hour of compressor (Item 125 ÷ Item 129).....	lb.
138. Steam less water (or superheated steam) consumed per ton of refrigeration at intake pressure and temperature (Item 124 ÷ Item 122a).....	lb.
139. Equivalent steam consumed per standard ton of refrigeration at intake pressure and temperature (Item 125 ÷ Item 122a).....	lb.

TABLE XXXIV.—ADDITIONAL ITEMS APPLYING ONLY WHEN THE DRIVING ELEMENT IS AN INTERNAL-COMBUSTION ENGINE

Description, Dimensions, Etc., of Compressor Drive

47. Type of engine; two or four cycle, single or double acting, horizontal or vertical
48. Auxiliaries attached, such as magneto, fuel-injection pump, fuel-circulating pump, lubricating-oil pumps, jacket circulating pumps, scavenging pumps, spray-air compressor, maneuvering or starting air compressor, radiator fans, oil or fuel coolers or heaters.....
49. Auxiliaries, independent or separately driven, and power.....
50. Grade of fuel for which engine is designed, kind of gas or specification of liquid fuel, and what was used in test.....
51. Diameter of all cylinders working, four cycle or two cycle.....in.
52. Stroke of working cylinders.....ft.
53. Diameter of piston and tail rods of working cylinders.....in.
54. Head-end horsepower constant for working cylinders (stroke $\times$ net piston area $\div$ 33,000).....
55. Crank-end horsepower constant for working cylinders (stroke $\times$ net piston area $\div$ 33,000).....

Power Input

58. Indicated horsepower of all working cylinders.....i.hp.
59. Indicated horsepower of each single-acting cylinder or each end of each double-acting four-cycle.....i.hp.

Test Data and Results

Pressures, Average

72. Spray-air pressure (air-injection Diesel engines) average (say whether gage or absolute).....lb. per sq. in.
73. Exhaust back pressure.....in. of water
74. Gas pressure in main gaseous fuel...in. water.....lb. per sq. in.
75. Scavenging air pressure average, two-cycle engines (say whether gage or absolute).....lb. per sq. in.

Temperatures

102. Temperature of fuel.....°F.
103. Temperature of main air supply.....°F.
Location of thermometer.....
104. Temperature of main jacket water (or oil) inlet.....°F.
105. Temperature of main jacket water (or oil) outlet.....°F.
106. Temperature of mixture at intake port (carbureter engine).....°F.

Fuel Properties

56. Heating value of gas, high value.....B.t.u. per cu. ft.
57. Heating value of liquid fuel, high value.....B.t.u. per lb.

Total Quantities

109. Total gaseous fuel at meter pressure and temperature.....cu. ft.
110. Correction factor for gas (see Test Code for Internal-Combustion Engines)....
111. Total gaseous fuel at carbureter or atmospheric pressure.....cu. ft.
112. Total liquid fuel.....lb.

Unit Quantities

113. Total gaseous fuel per hour at atmospheric pressure and temperature..cu. ft.
 113a. Total liquid fuel per hour.....lb.
 114. Total heat in fuel supplied per hour (Item 113 $\times$ Item 56 or Item 57)....B.t.u.

Economy Results

133. Fuel consumption per indicated horsepower-hour (Item 113 $\div$ Item 58).....lb. or cu. ft.
 134. Heat consumption per indicated horsepower-hour (Item 114 $\div$ Item 58)..B.t.u.
 135. Fuel consumption per standard ton of refrigeration (Item 113 $\div$ Item 122a)...
 136. Heat consumption per standard ton of refrigeration (Item 114 $\div$ Item 122a)...

A.S.M.E. TEST CODE (1926) FOR ABSORPTION REFRIGERATING SYSTEMS

TABLE XXXV.—DATA AND RESULTS OF TEST ON ABSORPTION REFRIGERATING SYSTEMS

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

Description, Dimensions, Etc.

7. Type of condenser.....
 8. Area of condenser surface on refrigerant side.....sq. ft.
 9. Type of evaporator.....
 10. Area of evaporator surface on refrigerant side.....sq. ft.
 11. Type of water pumps.....
 12. Rated capacity of water pumps.....gal. per min.
 13. Type of brine pumps.....
 14. Rated capacity of brine pumps.....gal. per min.
 15. Type of cooling tower.....
 16. Rated capacity of cooling tower.....
 17. Type of absorber.....
 18. Area of absorber surface on refrigerant side.....sq. ft.
 19. Type of generator.....
 20. Area of heating surface in generator.....sq. ft.
 21. Type of strong-liquor pump.....
 22. Diameter of piston rod, liquor pump.....in.
 23. Stroke of liquor pump.....in.
 24. Piston displacement of liquor pump.....cu. ft.
 25. Type of rectifier.....
 26. Cooling surface in rectifier.....sq. ft.
 27. Type of exchanger.....
 28. Cooling surface in exchanger.....sq. ft.

General Data

29. Composition of brine.....
 30. Specific gravity of brine compared with water at 39.1°F.....
 31. Specific heat of brine.....
 32. Specific gravity of liquid refrigerant as measured.....
 33. Concentration of strong liquor.....
 34. Concentration of weak liquor.....

35. Quality of steam to generator.....	
36. Kind of cooling water used.....	
37. Specific gravity of cooling water.....	
38. Specific heat of cooling water.....	

Test Data and Results

39. Duration of tests.....	hr.
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Average Pressures

40. Barometric pressure.....	in. of mercury at 32°F.
(a) Corresponding absolute pressure.....	lb. per sq. in.
41. Pressure in evaporator, by gage.....	lb. per sq. in.
42. Steam pressure in generator, by gage.....	lb. per sq. in.
43. Pressure in absorber, by gage.....	lb. per sq. in.
44. Pressure in condenser, by gage.....	lb. per sq. in.

Average Temperatures

45. Engine-room temperature.....	°F.
46. Condenser-room temperature.....	°F.
47. Temperature of outside air.....	°F.
48. Temperature of wet bulb.....	°F.
49. Temperature of liquid from condensers.....	°F.
50. Temperature of liquid from liquid receivers.....	°F.
51. Temperature of liquid from liquid cooler.....	°F.
52. Temperature of liquid at measuring point.....	°F.
53. Temperature of water to cooling tower at measuring point.....	°F.
54. Temperature of water from cooling tower at measuring point.....	°F.
55. Temperature of brine, brine-cooler inlet at measuring point.....	°F.
56. Temperature of brine, brine-cooler outlet at measuring point.....	°F.
57. Temperature range through brine coolers by electric-resistance thermometer.....	°F.
58. Temperature of cooling water, inlet to rectifier.....	°F.
59. Temperature of cooling water, outlet to rectifier.....	°F.
60. Temperature of weak liquor to weak-liquor cooler.....	°F.
61. Temperature of weak liquor from weak-liquor cooler.....	°F.
62. Temperature of water, inlet to weak-liquor cooler.....	°F.
63. Temperature of water, outlet from weak-liquor cooler.....	°F.
64. Temperature of weak liquor, inlet to absorber.....	°F.
65. Temperature of weak liquor, exit to generator.....	°F.
66. Temperature of strong liquor, inlet to exchanger.....	°F.
67. Temperature of strong liquor, outlet from exchanger.....	°F.
68. Temperature of condenser cooling water, inlet.....	°F.
69. Temperature of condenser cooling water, outlet.....	°F.
70. Temperature of steam to generator.....	°F.
71. Temperature of condensed steam from generator.....	°F.

Total Quantities

72. Liquid delivered to evaporator.....	lb.
73. Brine circulated.....	lb.
74. Strong liquor.....	lb.
75. Weak liquor.....	lb.
76. Steam condensed in generator.....	lb.

Unit Quantities

77. Liquid delivered to evaporator.....lb. per min.
 78. Brine circulated.....gal. per min.
 79. Cooling water supplied to condensers.....gal. per min.
 80. Cooling water supplied to absorber.....gal. per min.
 81. Cooling water supplied to rectifier.....gal. per min.
 82. Cooling water supplied to weak-liquor cooler.....gal. per min.

Refrigerating Output

83. Available cooling effect per hour (Q_e) (pp. 469 and 473).....B.t.u.
 84. Standard commercial tons of refrigeration produced (Item 83 $\div$ 12,000)....tons
 85. Standard commercial tons of refrigeration produced, brine method as a check:
 (Item 78 $\times$ Item 57 $\times$ Item 30 $\times$ 8.33 $\times$ Item 31 $\div$ 200).....tons

Economy Results

86. Tons of refrigeration per pound steam used in generator.....tons

Generator

87. Heat imparted by steam per minute.....B.t.u.
 88. Heat imparted by steam per hour (Item 87 $\times$ 60).....B.t.u.
 89. Logarithmic mean temperature difference (see Items 94 and 98).....°F.
 90. Coefficient of heat transfer: $K = \text{Item 88} \div (\text{Item 89} \times \text{Item 20})$ B.t.u.

Condenser

91. Heat removed per minute by condensing water (Item 69 — Item 68)
 $\times$ Item 38 $\times$ Item 79 $\times$ 8.33 $\times$ Item 37.....B.t.u.
 92. Heat removed as calculated from refrigerant condenser as a check (page
 472).....B.t.u.
 93. Heat removed per hour (Item 91 $\times$ 60).....B.t.u.
 94. Logarithmic mean temperature difference.....°F.
 95. Coefficient of heat transfer: $K = \text{Item 93} \div (\text{Item 94} \times \text{Item 8})$ B.t.u.

Evaporator

96. Heat removed per minute from brine.....B.t.u.
 97. Heat removed per hour (Item 96 $\times$ 60).....B.t.u.
 98. Logarithmic mean temperature difference.....°F.
 99. Coefficient of heat transfer: $K = \text{Item 97} \div (\text{Item 98} \times \text{Item 10})$

Absorber

100. Heat removed by water per minute.....B.t.u.
 101. Heat removed by water per hour (Item 100 $\times$ 60).....B.t.u.
 102. Logarithmic mean temperature difference.....°F.
 103. Coefficient of heat transfer: $K = \text{Item 101} \div (\text{Item 102} \times \text{Item 18})$ B.t.u.

Rectifier

104. Heat removed per minute (Item 59 — Item 58) $\times$ Item 38 $\times$ 8.33 $\times$ Item
 81 $\times$ Item 37.....B.t.u.
 105. Heat removed per hour (Item 104 $\times$ 60).....B.t.u.

Weak-liquor cooler

106. Heat removed per minute (Item 63 — Item 62) $\times$ Item 38 $\times$ 8.33 $\times$ Item
 82 $\times$ Item 37.....B.t.u.
 107. Heat removed per hour (Item 106 $\times$ 60).....B.t.u.

Strong-liquor Pump

108. Indicated horsepower.....i.hp.
 109. Steam consumption.....lb. per hr.

Brine Pumps

110. Gross or brake horsepower.....b.hp.
 111. Water horsepower.....hp.
 112. Overall efficiency.....per cent

Water Pumps

113. Gross or brake horsepower.....b.hp.
 114. Water horsepower.....hp.
 115. Overall efficiency.....per cent

CHAPTER XIX

TESTING OF CONDENSERS AND FEED-WATER HEATERS

A.S.M.E. TEST CODE (1926) FOR CONDENSING APPARATUS (ABRIDGED)

It should be borne in mind at the very beginning of any investigation of condensing machinery that, while some classes of condensers are apparently simple pieces of apparatus, a complete test of such a device as a surface condenser represents a greater complement of diversified auxiliary apparatus than any other unit in the power plant. Consequently, a complete test requires the test of the condenser itself and, as conditions demand, either complete or partial tests of the auxiliaries. This code has been framed to include all types of condensing equipment with their auxiliaries, but such items as have no bearing on the object in view can, of course, be omitted.¹ The various classes of condensing apparatus which it covers and the principal auxiliary pumping equipment concerned are grouped as follows:

Surface condensing:

- (a) Surface condenser. (b) Circulating pump, steam driven or motor driven.
- (c) Air pump, either reciprocating dry-vacuum or hydraulic vacuum pump, either steam driven or motor driven, or steam-air ejector. (d) Condensate pump, steam driven or motor driven.

Low level jet condensing and barometric condensing:

- (a) Jet condenser. (b) Supply or tail pump, steam driven or motor driven.
- (c) Air pump, either reciprocating dry-vacuum or hydraulic vacuum pump, either steam driven or motor driven, or steam-air ejector.

Ejector condensing:

- (a) Condenser. (b) Supply pump, steam driven or motor driven.

Instruments and apparatus required for a condenser test consist of the following:

- a. Barometer, preferably of the mercurial type.
- b. Mercury columns for measuring vacuums having the smallest scale graduations not greater than 0.1 inch.
- c. Bourdon gages for measuring pressures when too great for mercury manometer tubes.

¹ As there is often a difference in the pressure maintained in the various parts of the apparatus, the condenser guarantee usually specifies a definite absolute pressure to be maintained at the steam entrance to the condenser, when condensing a stated amount of steam in pounds per hour, the condenser being supplied with a sufficient amount of circulating water (usually specified) at a stated temperature, it being further understood that the condenser, prime mover, and connections shall be free from air leaks. It is sometimes stated that the air to be removed by the vacuum pump shall not exceed a certain figure.

d. Thermometers: (1) For registering temperatures of circulating water, condensed steam (condensate), and vapors—graduated by half degrees and with maximum scale readings from 32 to 150°F.; (2) for determining steam temperature graduated by degrees and with maximum scale readings from 32 to 250°F.

e. Tanks and platform scales for measuring water (or water meters calibrated in place under conditions of use).

f. Appliances for tests of auxiliary pumps, such as steam-engine indicators, electrical instruments, speed indicators, tachometers.

g. Air-measuring devices—gasometers or calibrated orifices (page 189).

h. The same apparatus may be employed for measuring the quantity of steam condensed as for testing the engine, turbine, or other steam machinery supplying the condenser.

i. For measuring circulating or injection water, weirs, nozzles, orifices, and Pitot tubes may be used or the amount of circulating water may be estimated by the heat-balance method as given by the following equation:

$$C = \frac{H_c W_c}{t_o - t_i},$$

in which C = pounds of circulating water per hour.

H_c = B.t.u. per pound of steam absorbed by the condenser (page 491).

W_c = steam discharged to the condenser, in pounds per hour.

t_o = temperature of circulating water going out, degrees Fahrenheit.

t_i = temperature of circulating water going in, degrees Fahrenheit.

If the mercury columns show serious discrepancies, the situation must be specially studied. Possibly more columns may be necessary.

The readings of all gages, corrected to inches of mercury at 32°F. shall be averaged and subtracted from a similarly corrected reading of a mercurial barometer to give the *condenser inlet pressure*. Correction must be made for the difference in elevation of the barometer and the vacuum gages, if any. Inlet pressure shall not be computed from temperature observations. The barometer and vacuum gage should be located near together, to avoid the necessity of correcting for altitude (page 7) and temperature. The temperature correction may be made negligibly small by maintaining mercury columns and barometer at approximately the same temperature; but if this is done, the temperature correction shall be computed for sufficient cases to demonstrate that it is negligible.

The performance of the condenser is directly influenced by the amount of air leakage which varies greatly in actual service and should be brought down to the quantity recognized as permissible in the size and type of prime mover under test. The quantity of air removed from the condenser shall be measured by a gasometer or other approved apparatus.

If the object of the test permits, any serious leakages, whether of air, steam, or water, shall be rectified before proceeding with the main trial.

In locating thermometers within the steam space of a condenser, care should be observed that the thermometer wells in which they are placed are shielded from direct impact of steam currents and from radiation from the cold tubes. In most cases it is desirable to insert bare thermometers through rubber stoppers (page 267). Thermometers

showing the temperature of discharge water should be located as far as practicable, but not farther than 100 feet from the condenser, to allow thorough mixing of the water; otherwise, especially with large pipes and with special forms of circulation, the temperature will vary considerably in different sections. Not less than three thermometers usually placed 120 degrees (one-third of circle) apart should be used for determining the temperature of the discharge circulating water.

Correct application and use of the mercury gage, preferably a full-length mercury column, for determining vacuums are all-important in a condenser test. Long pipe connections should not be used. To avoid capillarity, the connecting pipes should be not less than $\frac{1}{2}$ -inch iron-pipe size, and they should be arranged to pitch toward the inlet conduit connection and contain no pockets. The pressure at the steam inlet to the condenser shall be measured by means of at least two carefully constructed mercury columns, accurate to 0.01 inch of mercury, used with all precautions and corrected for all observable causes of error. Each column shall be mounted as near as is practicable to the pressure hole in the conduit or casing, and so as to be as nearly as possible free from vibration. It shall be connected to the inlet passage of the condenser, through pressure holes located within the condenser inlet nozzle.¹

If the conduit connecting the turbine exhaust nozzle and the condenser inlet nozzle is approximately straight, the inlet pressure shall be measured as far as possible from the turbine exhaust flange (toward the condenser), and at a distance from the condenser inlet flange (toward the turbine) not to exceed three diameters of a circle of area equal to that of the inlet conduit. Each mercury column shall be connected to a static pressure hole normal to the inner wall of the conduit, and not smaller than $\frac{1}{2}$ -inch standard iron-pipe size. There shall be approximately one such hole and column for each 16 square feet of inlet-conduit area, but in no ordinary case less than two or more than six columns. The pressure holes shall be distributed as uniformly as possible around the periphery of the conduit. When both turbine (or engine) and condenser are tested together, in many cases common readings of exhaust pressure will suffice for both tests.

If there is *no straight conduit* between turbine and condenser, measurement shall be made in exactly the same way, but the pressure holes shall be located within 12 inches of the plane of the face of the condenser inlet flange, and at that plane where the steam flow is most likely

¹ Extreme care should be exercised to see that the piping and joints between a mercury column and its pressure hole are airtight. One way is to place a valve with the stuffing box on the gage side of the pressure line, immediately adjacent to the connection to the inlet conduit. At intervals during the condenser tests, while vacuum is on, the valve is to be closed. If the mercury column reading does not change at a rate greater than 1 inch in 2 minutes, the pressure line may be assumed to be tight.

to be smooth. If both turbine (or engine) and condenser are tested together, in this case, two sets of readings will in general be required.

In testing large condensers, if it is unduly difficult or considered undesirable to put two or more flush holes in the inlet connection as specified above, it shall be optional with the parties concerned, upon previous written agreement, to install other types of pressure tips with appropriate precautions. Such tips must be of a type of demonstrated accuracy. At least two such pressure tips shall be used, one located centrally in each half of the inlet opening and within 12 inches of the plane of the face of the condenser inlet flange.

Condenser Leakage.—The quantity of leakage from the water spaces to the steam space in the case of surface condensers may be determined by measuring the water discharged from the condensate pump when running dry, that is, with all steam shut off from entering the condenser, the normal quantity of circulating water being in use and other conditions being normal. The quantity of leakage should be determined for at least an hour's time, after normal conditions are established, and deducted from the total condensate as measured on the main test. It is recommended that, if this leakage exceeds 1 per cent of the normal capacity of the condenser, no further test be made until such leakage is reduced to the stated figure. The average of such a test made before and after the main test may be considered as the condenser leakage, but attention is called to the fact that under operating conditions the actual leakage may vary from that obtained during the leakage test, owing to the different temperature conditions and vibrations that may exist. This fact emphasizes the necessity for *reducing the leakage to the absolute minimum before the test*.

Other methods of determining the amount of leakage have been successfully applied but are not universally applicable, as, for example, the *electrolytic method*, (page 355) which is based on the change in electrical resistance of condensate due to contamination with salt-water leakage. Also the *chemical method of titration* with a solution of silver nitrate (page 347), with potassium chromate used as an indicator.

The tightness of the condenser and its connections with respect to air leakage, whatever the type of condenser, should be carefully ascertained. The surface condenser shell and its connections should be filled with water at a temperature approximately that of the surrounding air and under some slight pressure, and the leaks revealed. Care should be taken when filling a condenser with water to see that the condenser supports are not overloaded and an undue load is placed on the exhaust nozzle of the turbine or engine, in cases where the condenser is mounted on springs and there is no expansion joint between the turbine or engine and the condenser. A convenient method of determining the *fact* of air leakage, though not its degree, is to shut off the injection water, stop the admission of steam, stop the vacuum pump, and then observe how long a time the vacuum holds. If the vacuum falls at once, a thorough search by the aid of a candle flame will serve to locate the leakage.

Starting, Stopping, and Duration of Tests.—The starting, stopping, and duration of a condenser test should follow the same rules as those

governing a steam-engine or turbine test, and reference should be made to the Steam Engine Code and Steam Turbine Code for the general directions to be followed.

In *surface-condenser* tests the quantity of steam condensed should be determined by collecting and weighing the condensate, and correcting the total, as measured, for leakage, if any.

In *jet condensers*, including those of the barometric type:

The quantity of steam condensed may be ascertained by measuring the feed-water consumption and correcting for drips, for leakage of boilers and piping, and for steam which is not supplied to the condenser, preferably by taking steam from an isolated group of boilers.

It may be determined approximately by steam-meter measurement if the meter is calibrated in place and under conditions of use; or by calculation from the heat units given up to the injection water, if determined.

When the circulating water is discharged into an open tank, a notch weir (page 216) or a rectangular weir may be arranged for the measurement. Otherwise an orifice may be inserted in the discharge pipe and orifice measurement (page 214) employed. Any of these indirect methods must be considered as furnishing approximate measurements, especially when the weir or orifice is in a limited space and it is impracticable to calibrate it in place or under its working conditions. The calculation is also affected by the accuracy with which temperatures are read as well as the correctness of installation and the fineness of graduation of the thermometers. This becomes of great importance in cases of small temperature rise.

The quantity of steam consumed by auxiliaries may be found by connecting them separately or collectively to a surface condenser, if available, and measuring the condensate.

Where jet, barometric, or ejector condensers are used, any air leakage into the water-supply line is as serious as a direct leak into the prime mover or condenser. Generally, a portion of the water-suction line is under partial vacuum owing to the fact that either the condenser itself or the circulating pump draws water from the injection source, which is usually at a lower elevation than the condenser or its supply pump. A convenient method of determining whether there is any infiltration of air into this line is to place a gage or mercury column directly beneath the valve on the injection line, which is usually located immediately adjacent to the condenser line, and observing the drop in the vacuum reading after the valve has been shut and the circulating pump shut down. A drop in the vacuum reading will either indicate air leakage into that part of the line under vacuum or a leak in the valve itself.

Heat Supplied to Condenser per Hour.—The size of the condensing equipment should be determined by the amount of work it is to perform—that is, the amount of heat it is to remove. This depends upon the heat contents of the steam as delivered to the condenser, an amount seldom specified, although of prime importance. The heat supplied to the condenser is that supplied to the prime mover, exhausting to the condenser, less that converted into work and lost by generator losses, friction, radiation, and other wastes, not readily determined. It is simply expressed by the following formulas (where the condenser is bolted direct to the turbine exhaust opening):

For a simple nonbleeding¹ and nonreheating² turbine the heat supplied is,

¹ A **nonbleeding steam turbine** is one which has no provision for removing steam that is not returned from any part of the turbine casing except at the exhaust connection (see "Steam Turbines," by the author, 6th ed., p. 369).

² A **nonreheating steam turbine** is one which has no provision for removing steam from some point in the turbine casing for the purpose of passing it through a heating

$$h_{gs} = h_{gt} - \frac{3,413^*}{S \times E}$$

For a reheat, nonbleeder turbine, $h_{gs} = h_{gt} + (h_{gr} - h_{gz}) - \frac{3,413}{S \times E}$ in which the expression $(h_{gr} - h_{gz})$ is repeated for each stage of reheating.

For a bleeder, nonreheat turbine,

$$h_{gs} = \frac{h_{gt}W}{W - (w_1 + w_2 + \dots w_n)} - \frac{3,413 \text{ kw}}{E[W - (w_1 + w_2 + \dots w_n)]} - \frac{w_1 h_{g1} + w_2 h_{g2} + \dots w_n h_{gn}}{W - (w_1 + w_2 + \dots w_n)}$$

For a bleeder and reheat turbine,

$$H_{gs} = \frac{H_{gt}W}{W - (w_1 + w_2 + \dots w_n)} + \frac{W - w(h_{gr} - h_{gz})}{W - (w_1 + w_2 + \dots w_n)} - \frac{3,413 \text{ kw}}{E[W - (w_1 + w_2 + \dots w_n)]} - \frac{w_1 h_{g1} + w_2 h_{g2} + \dots w_n h_{gn}}{W - (w_1 + w_2 + \dots w_n)}$$

In the equations given above

h_{gs} = B.t.u. per pound of steam at the prime-mover exhaust.

h_{gt} = B.t.u. per pound of steam at the prime-mover throttle.

S = steam supplied to throttle of prime mover, in pounds per kilowatt-hour.

E = mechanical and electrical efficiency of prime mover complete, including all friction, electrical loss, windage, radiation, etc.

h_{gr} = B.t.u. per pound of steam at point of reentrance to turbine after reheating.

h_{gz} = B.t.u. per pound of steam after partial expansion in the turbine at point of extraction for reheat.

W = total steam supplied to throttle of prime mover, in pounds per hour.

$w_1, w_2 \dots w_n$ = steam bled from first, second, and n th bleeder points of turbine, in pounds per hour.

$h_{g1}, h_{g2} \dots h_{gn}$ = B.t.u. per pound of steam at the first, second, and n th bleeder points of the turbine.

kw = load on prime mover, in kilowatts.

In using the formulas given above for determining the *heat absorbed by the condenser*, the heat carried away by the condensate should be subtracted from H_c , or heat absorbed by condenser.

$$h_{gc} = h_{gs} - (t_c - 32),$$

in which h_{gc} = B.t.u. per pound of exhaust steam absorbed by condenser and carried away by the circulating water and t_c = temperature of condensate in degrees Fahrenheit.

If the prime-mover steam rate is given in pounds per horsepower-hour, the constant, 2,545, must be substituted for 3,413. The value of E varies with both the type and the size of the prime mover.

A check upon the quantity of heat so determined is obtained by multiplying the weight of circulating or injection water in pounds per hour by its rise in temperature in its passage through the condenser—but such readings as already explained can be considered as only approximate.

device to increase the total heat in the steam and then returning this steam to the stage of the turbine from which it was withdrawn.

* The heat equivalent of 1 kilowatt-hour is 3,413 B.t.u. per hour; and 1 horsepower-hour is equivalent to 2,545 B.t.u. per hour.

Determination of Heat Transfer in Surface Condenser.—The efficiency of the surface condenser is expressed in B.t.u. transmitted per square foot of the tube surface per hour per degree mean difference in temperature between the two sides of the tube. Both the *arithmetical* (page 577) and *logarithmic mean difference* (page 472) are used for this purpose and it is recommended that the test results be given for both in order to make comparison possible with all other tests.

A complete comparative discussion here is not desirable, but it is permissible to state that the determination of both is based upon assumptions which do not correctly represent the conditions, in practice.

The formulas follow:

$$(a) \text{ Arithmetical mean difference} = t_s - \frac{t_o + t_i}{2};$$

$$(b) \text{ Logarithmic mean difference} = \frac{t_o - t_i}{\log_e \frac{t_s - t_i}{t_s - t_o}};$$

in which t_s = temperature in steam space.

t_i = temperature of incoming circulating water.

t_o = temperature of outgoing circulating water.

The main points of difference are as follows: The arithmetical mean difference (a) assumes: (1) a constant steam temperature throughout the condenser and (2) a uniform rise in temperature of the circulating water from its entrance to exit. The logarithmic mean difference (b) is based upon: (3) a constant steam temperature throughout the condenser and (4) a uniform heat transfer throughout the condenser.¹

The transfer of heat through a tube surface from air to water has been given by different experimenters as varying from 6 to 10 B.t.u. per square foot per degree Fahrenheit per hour, or less than 1 per cent of that from steam to water. If we have 180,000 pounds of steam per hour entering a surface condenser and carrying with it, say, 8 cubic feet of free air per minute, or 36 pounds per hour, we shall have at the entrance 5,000 pounds of steam per pound of air, the latter being negligible, and the temperature of steam as indicated by a correctly placed thermometer will be practically that corresponding to the absolute pressure at that point.

It will oftentimes be found in practice that the curve of circulating-water temperature rise will more nearly conform to a straight-line performance. Some authorities have advised the use of the arithmetical mean as sufficiently accurate for purposes of comparison, especially when the rise in temperature of circulating water is small. Plotted curves will show a difference of from 3 to 10 per cent in the mean difference, as derived for usual practical conditions at 1½ to 2 inches of mercury, absolute pressure, and, say, 20 per cent around 3 inches.

Most engineers prefer the *logarithmic mean*, which in any event should be adhered to for all heater calculations where the range of temperature is high. The logarithmic mean m of varying temperatures may be defined in terms of d_1 and d_2 the temperature differences at the two ends of the *condenser tubes*, by the following formula:

$$m = (d_1 - d_2) \div 2.3 \log_{10} (d_1 \div d_2).$$

A.S.M.E. TEST CODE (1926) FOR CONDENSING APPARATUS

TABLE XXXVI.—DATA AND RESULTS OF CONDENSING-APPARATUS TESTS

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test.

¹ Numerous tests show that the temperature is not uniform throughout a condenser, and that the rate of heat transfer is not uniform.

Description and Dimensions, Etc.

7. Unit number.....Manufacturer's Serial No.....
 Drawing No.....
 How long in service?.....
 If surface type, how long since tubes were last cleaned?.....
8. Condensing steam from—No.—size—make—type (horizontal or vertical)
 turbine or engine.....
9. Type of condenser (horizontal or vertical—jet or surface).
 (a) Material of shell.....
 (b) Weight of condenser, empty.....lb.
 (c) Weight of condenser, including circulating water.....lb.
 (d) Shape and approximate overall dimensions.....
 (e) Vacuum breaker—type and description.....
10. Distance between exhaust flange of prime mover and condenser exhaust opening,
 size of connecting pipe, with enumeration and description of fittings and valves
 in same.....
 Are valve disks, stems and joints water-sealed?.....
11. Exhaust relief valves: size, make, and location.....
 Is relief valve water-sealed?.....
12. Source of supply of circulating or injection water (salt or otherwise)
 (a) Analysis of condensing water.....

Surface Condenser

[Whether (1) standard type, water inside tubes, or (2) water-works type, water outside tubes.]

13. Number of passes (water if 1) (steam if 2).
14. Amount of effective *cooling* surface in square feet (including air cooler but excluding internal heater) measured on the outside of the tubes between tube heads if (1) on the inside, if (2) tube-head area not included.
15. Amount of effective *heating* surface in internal heater, if any—square feet (see drawings).
16. Location and description of internal heater, if any.
17. Amount of effective cooling surface in air cooler.....
18. Location and description of air cooler (see drawings).....
19. Diameter, gage, material, and length of tubes between heads.....
20. Number of tubes in passes: (a) tubes in first pass; (b) tubes in second pass; (c) tubes in third pass; (d) tubes in fourth pass.
21. Dimensions, shape, and material of tube head and total area in square feet of each tube head, in steam space (before drilling for tubes).
22. Number, material, and total thickness of support plates.....
23. Per cent of tube-head area in contact with steam.....
24. Per cent of volume of shell actually occupied by tubes below a horizontal line at top of tube head—inside shell.....
25. Description or sketch of tube layout.....
26. Kind of tube packing.....
27. Condition of tube surface (as to cleanliness).....

Low-head Jet, Barometric, or Ejector Condenser—Parallel-flow or Countercurrent

28. Approximate volume of space available for condensation.....cu. ft.
29. Type of water distributor (jet or gravity).....
30. Number and type of jets.....
31. Mechanical condition of apparatus.....

Special Features

32. Description of special features including internal or external cooler, if any.

Circulating Pump of Surface Condenser, or Removal (or Tail) Pump of Low-head Jet Condenser, or Injection Pump of Barometric or Ejector Condenser

33. Manufacturer, serial number, and drawing number.
 34. Type, horizontal or vertical—centrifugal or reciprocating.
 35. Number of stages if centrifugal—single or duplex—single or double suction.
 36. How driven (engine, turbine, or motor).
 37. Rated horsepower, dimensions of cylinders, and stroke of reciprocating apparatus
 38. Normal r.p.m. and characteristics of motor.
 39. Size of all pipe openings.
 40. State special features, type of valves, direct gear or belt driven.
 41. Manufacturer's guarantee, if any.
 42. Mechanical condition.

Air-vapor-removal Apparatus**Reciprocating dry vacuum pump:**

43. Manufacturer, serial number, and drawing number.
 44. Type (horizontal or vertical) number of stages.
 45. How driven (steam cylinder or motor).
 46. Size:
 (a) Diameter steam cylinder and stroke. in.
 (b) Diameter air cylinder first stage. in.
 (c) Diameter air cylinder second stage. in.
 (d) Diameter of piston rods—air cylinders. in.
 (e) Diameter of piston rods—steam cylinders. in.
 47. Type and description of air valves.
 48. Actual air piston displacement per revolution. cu. ft.
 49. Clearance in per cent of air piston displacement.
 50. Type, size, and description of motor (if motor driven).
 51. Special features.
 52. Manufacturer's guarantee, if any.
 53. Mechanical condition.

Hydraulic type:

54. Manufacturer, serial number, and drawing number.
 55. Type and trade name.
 56. Size including supply pump if separate.
 57. How driven (engine, turbine, or motor).
 58. Give indicated horsepower or brake horsepower.
 59. Size of all pipe openings.
 60. Special features.
 61. Manufacturer's guarantees, if any.

Steam-jet or ejector type:

62. Manufacturer, serial number, and drawing number.
 63. Type, trade name, number of stages.
 64. Number and sizes of precoolers, intercoolers, and aftercoolers (jet or surface) and
 amount of surface in each.
 65. Size of steam and water connections.
 66. Special features including supply of water for coolers, removal of condensate, etc.
 67. Are single or multinozzles used in ejectors?

- 68. Manufacturer's guarantees, if any.....
- 69. Mechanical condition.....

Condensate pump:

- 70. Manufacturer, serial number, and drawing number.....
- 71. Type—horizontal or vertical—number of stages.....
- 72. Size of suction and discharge openings.....
- 73. How driven (engine, turbine, or motor) brake horsepower and speed.....
- 74. Special features.....
- 75. Manufacturer's guarantees, if any.....
- 76. Mechanical condition.....

Test Data and Results

- 77. Duration of test.....hr.

Steam Supply

- 78. Load on prime mover or other apparatus exhausting to condenser. i.hp. or kw.
- 79. Heat equivalent of work per hour.....B.t.u.
- 80. Absolute steam pressure at main throttle.....lb. per sq. in.
- 81. Quality of steam at main throttle.....per cent moisture or deg. superheat
- 82. Quantity of steam entering condenser.....lb.
- 83. Quantity of steam at throttle of each auxiliary if required.....lb.

Average Pressures and Temperatures

- 84. Barometric pressure as observed.....in. of mercury
 - (a) Temperature.....°F.
- 85. Barometric pressure as corrected.....in. of mercury at 32°F.
- 86. Temperature of atmospheric air.....°F.
 - (a) Relative humidity.....per cent
- 87. Absolute pressure of steam leaving prime mover or other
 - apparatus at exhaust flange.....in. of mercury at 32°F.
- 88. Absolute pressure of steam entering condenser or other
 - apparatus at inlet flange.....in. of mercury at 32°F.
- 89. Saturated-steam temperature corresponding to the pressure in Item 88.....°F.
- 90. Measured temperature corresponding to Item 89.....°F.
- 91. Absolute pressure at air-removal flange.....in. of mercury at 32°F.
- 92. Absolute pressure at air-removal pump suction nozzle. in. of mercury at 32°F.

Surface Condenser

- 93. Total steam delivered to prime mover or other apparatus exhausting to condenser as measured.....lb.
 - (a) Rate per hour.....lb.
- 94. Total steam bled from first bleeder connection.....lb.
 - (a) Rate per hour.....lb.
- 95. Total steam bled from second bleeder connection.....lb.
 - (a) Rate per hour.....lb.
- 96. Total steam bled from *n*th bleeder connection.....lb.
 - (a) Rate per hour.....lb.
- 97. Total water delivered by hot-well pump.....lb.
 - (a) Rate per hour.....lb.
- 98. Water leakage including steam from seals per hour.....lb.
 - (a) Immediately prior to test.....lb.
 - (b) Immediately after test.....lb.
 - (c) Average leakage—corrected to rate per hour.....lb.

99. Prime-mover steam condensed by condenser.....lb.
 (a) Rate per hour.....lb.
100. Temperature of circulating water
 (a) Entering condenser.....°F.
 (b) Leaving first pass.....°F.
 (c) Leaving second pass.....°F.
 (d) Leaving third pass.....°F.
 (e) Leaving condenser.....°F.
101. Temperature of condensate in hot well.....°F.
102. Temperature of condensate leaving internal heater—if any.....°F.
103. If make-up or other foreign water is introduced to condenser or hot well, additional data should be furnished as to quantity, temperature, etc.

Circulating Pump

104. Circulating water (measured if possible).....gal. per min.
105. Circulating-water pressure above or below atmospheric
 (a) At condenser-inlet nozzle.....lb. per sq. in.
 (b) At condenser-outlet nozzle.....lb. per sq. in.
 (c) Difference in level between circulating nozzles center lines.....ft.
 (d) Net condenser-circulating friction.....lb. per sq. in.
106. Piston speed or revolution per minute if centrifugal
107. (a) Pressure at pump intake nozzle, by gage.....lb. per sq. in.
 (b) Pressure at discharge, by gage.....lb. per sq. in.
 (c) Difference in level of gages.....lb. per sq. in.
 (d) Total corrected pumping head.....lb. per sq. in.
108. Absolute pressure of steam at throttle.....lb. per sq. in.
 (a) Quality of steam at throttle.....per cent moisture or °F. superheat
109. Absolute pressure at exhaust nozzle.....lb. per sq. in.
110. Total indicated horsepower or brake horsepower or electrical horsepower if motor-driven
111. Weight of steam consumed per hour.....lb.

Jet, Barometric, or Ejector Condensers

112. Amount of water handled by injection or removal pump.....gal. per min.
113. Temperature of injection water
 (a) Entering condenser.....°F.
 (b) Leaving condenser.....°F.
114. Absolute pressure of injection water in distributing box at inlet flange.....lb. per sq. in.
 (a) Height of same above center line of pump.....ft.
115. (a) Pump discharge pressure (by gage).....lb. per sq. in.
 (b) Pump intake pressure (by gage).....lb. per sq. in.
 (c) Difference in level between centers of gages.....ft.
 (d) Water level in condenser or hot well above center lines of pump.....ft.
116. Correct total pumping head.....lb. per sq. in.
117. Speed of pump.....r.p.m.

Air-vapor Removal Apparatus. All Types of Reciprocating Dry Vacuum Pumps—Steam or Motor Driven. All Types of Hydraulic Pumps—Steam or Motor Driven

118. Steam conditions:
 (a) Absolute pressure at throttle.....lb. per sq. in.
 (b) Quality of steam at throttle.....per cent moisture or °F. superheat

- (c) Absolute pressure at steam exhaust nozzle.....lb. per sq. in.
- (d) Weight of steam consumed per hour.....lb. per hr.
- 119. Air-vapor pressures and temperatures:
 - (a) Absolute pressure at pump suction nozzle.....in. of mercury at 32°F.
 - (b) Temperature air vapor at suction nozzle.....°F.
 - (c) Lowest absolute pressure obtainable with closed suction. (This determined before or at close of main test and with as near the same operating conditions as possible.).....in. of mercury at 32°F.
- 120. Horsepower of drivers:
 - (a) Horsepower of steam cylinders (if reciprocating).....i.hp.
 - (b) Brake horsepower of turbine.....b.hp.
 - (c) Power input if motor driven.....kw.
 - (1) Speed of motor.....r.p.m.
 - (2) Efficiency of motor.....per cent
- 121. Reciprocating dry vacuum pumps:
 - (a) Horsepower of vacuum cylinders.....i.hp.
 - (b) Speed.....r.p.m. or s.p.m.¹
 - (c) Displacement of piston.....cu. ft. per min.
 - (d) Mechanical efficiency of pump.....per cent
 - (e) Temperature of jacket cooling water entering.....°F.
 - (f) Temperature of jacket cooling water leaving.....°F.
 - (g) Amount of jacket cooling water.....lb. per hr.
 - (h) Temperature of air vapor at pump discharge nozzle.....°F.
- 122. Hydraulic type:
 - (a) Speed of pump.....r.p.m.
 - (b) Temperature of hurling water entering.....°F.
 - (c) Temperature of hurling water leaving.....°F.
 - (d) Amount of hurling water circulated.....gal. per min.
 - (e) Absolute pressure of hurling water at pump suction nozzle....lb. per sq. in.
 - (f) Absolute pressure of hurling water at discharge nozzle.....lb. per sq. in.
 - (g) Amount of hurling water make-up.....gal. per min.
 - (h) Temperature of hurling water make-up.....°F.
 - (i) Method of cooling hurling water, if this water is recirculated.....
- 123. Steam-air ejector:
 - (a) Absolute steam pressure at ejector throttle.....lb. per sq. in.
 - (b) Quality of steam at ejector throttle °F. superheat or per cent moisture
 - (c) Amount of steam used, first-stage nozzles.....lb. per hr.
 - (d) Amount of steam used, second-stage nozzles.....lb. per hr.
 - (e) Absolute air-vapor pressure at main suction nozzle..in. of mercury at 32°F.
 - (f) Temperature of air vapor at main suction nozzle.....°F.
 - (g) Absolute air-vapor pressure at suction nozzles, second stage.....in. of mercury at 32°F.
 - (h) Temperature of air vapor at suction nozzle, second stage.....°F.
 - (i) Temperature of water entering intercooler (if any—surface or jet).....°F.
 - (j) Temperature of water leaving intercooler.....°F.
 - (k) Amount of water used in intercooler.....gal. per min.
 - (l) Amount of condensate leaving intercooler (if surface).....lb. per hr.
 - (m) Temperature of condensate leaving intercooler.....°F.
 - (n) Absolute pressure at exhaust nozzle.....lb. per sq. in.
 - (o) Temperature at exhaust nozzle.....°F.

¹ Revolutions per minute or strokes per minute.

124. Actual amount of air discharged (as measured by gasometer or orifice (p. 189))
cu. ft. per min. at 60°F. and 14.7 lb. per sq. in. abs.

Condensate Pump

125. Speed of pump.....r.p.m.
 126. Heat contained in condensate above 32°F.....B.t.u.
 127. Total amount of water handled per hour.....lb.
 (a) Rate per minute.....gal. per min.
 128. Water pressures:
 (a) Pump discharge pressure (corrected to center line of pump).....ft.
 (b) Pump suction pressure (corrected to center line of pump).....ft.
 (c) Total head on pump.....ft.
 (d) Water level in condenser or hot well above center line of pump.....ft.
 129. Water horsepower from above.....hp.
 130. Power required to drive, if motor driven.....b.hp.
 131. Absolute pressure at throttle.....lb. per sq. in.
 132. Quality of steam.....per cent moisture or °F. superheat
 133. Absolute pressure at exhaust.....lb. per sq. in.
 134. Steam per hour required to drive (if steam driven).....lb.

Additional Data

135. Necessary additional data to cover operation of any special features or auxiliaries
 as referred to under Item 32.

General Results of Tests

136. Steam condensed per hour:
 (a) Guarantee.....lb.
 (b) Test [Item 99(a)].....lb.
 137. Absolute pressure maintained at condenser, exhaust nozzle:
 (a) Guarantee.....in. of mercury at 32°F.
 (b) Test (Item 88).....in. of mercury at 32°F.
 138. Temperature corresponding to test pressure (Item 89).....°F.
 139. Temperature of condensate in hot well:
 (a) Guarantee.....°F.
 (b) Test (Item 101).....°F.
 140. Temperature of entering condensing water:
 (a) Guarantee.....°F.
 (b) Test [Item 100(a)].....°F.
 141. Quantity of condensing water:
 (a) Guarantee.....gal. per min.
 (b) Test.....gal. per min.
 142. Amount of free air handled by air pump:
 (a) Guarantee.....cu. ft. per min. at 60°F. and 14.7 lb. per sq. in. abs.
 (b) Test (Item 124).....cu. ft. per min. at 60°F. and 14.7 lb. per sq. in. abs.
 143. Total heat delivery to prime mover:
 (From Items 78 to 82).....B.t.u. per hr.
 144. Total heat carried to condenser by exhaust steam
 (see pp. 490 and 491).....B.t.u. per hr.
 145. Total heat carried from condenser by condensate
 (From Items 97 to 101).....B.t.u. per hr.
 146. Total heat absorbed by condensing water
 (From Items 100 to 104).....B.t.u. per hr.

147. Steam supplied to auxiliaries:
 - (a) Absolute pressure at throttle.....lb. per sq. in.
 - (b) Quality of steam.....per cent moisture or °F. superheat
148. Steam consumed by auxiliaries:
 - (a) Guarantee.....lb. per hr.
 - (b) Test.....lb. per hr.
 - (c) Per cent of steam delivered to condenser.....per cent
149. Indicated horsepower or brake horsepower of auxiliaries:
 - (a) Guarantee.....hp.
 - (b) Test.....hp.
150. Mechanical efficiency of circulating or tail pump
(From Items 104 to 116).....per cent
151. Mechanical efficiency of condensate pump
(From Items 129 to 134).....per cent
152. Mechanical efficiency of air-removal pump.....per cent
153. Volumetric efficiency of air-removal pump.....per cent
154. Average velocity of circulating water through tubes.....ft. per sec.
155. Heat transfer per degree Fahrenheit difference of temperature per square foot per hour (based on total surface, Item 14):
 - (a) Using arithmetical mean.....B.t.u.
 - (b) Using logarithmic mean.....B.t.u.
156. Steam condensed per square foot of total surface per hour (Item 82 ÷ Item 14)
.....lb.
157. Circulating water per pound of steam condensed (Item 104 ÷ Item 99a)....lb.
158. Circulating water per square foot of total condenser surface (Item 104 ÷ Item 14).....lb.

TABLE XXXVII.—PRINCIPAL DATA AND RESULTS OF CONDENSER TEST

1. Dimensions of condenser and auxiliaries.....
2. Data regarding prime mover or other apparatus exhausting to condenser.....
3. Data regarding types and special features of all auxiliaries.....
4. Date of test.....
5. Steam condensed per hour.....lb.
6. Auxiliaries:
 - (a) Steam or power consumed by auxiliaries.....lb. or kw.
 - (b) Per cent of (a) to total steam or power of apparatus exhausting to condenser.....per cent
7. Absolute pressure at condenser exhaust nozzle.....lb. per sq. in.
8. Temperature of condensate in well.....°F.
9. Difference between temperature of steam entering condenser and temperature of water in hot well.....°F.
10. Circulating water:
 - (a) Quantity.....gal. per min.
 - (b) Temperature at inlet.....°F.
 - (c) Temperature at outlet.....°F.
11. Heat transfer per degree Fahrenheit difference of temperature per square foot of condensing surface per hour:
 - (a) Using arithmetical mean.....B.t.u.
 - (b) Using logarithmic mean.....B.t.u.
12. Volume of air removed per minute at 60°F. and 14.7 lb. per sq. in. abs....cu. ft.

TABLE XXXVIII.—SUPPLEMENTARY CODE

Operation Independent from Condenser or Shop Test

Air-vapor Removal Apparatus

1. Reciprocating Dry Vacuum Type:

- (a) Absolute steam pressure at throttle.....lb. per sq. in.
- (b) Quality steam at throttle °F. superheat or per cent moisture.....
- (c) Speed, in r.p.m. or strokes per minute (at which pump is designed to operate)
.....
- (d) Displacement of air piston.....cu. in.
- (e) With items (a), (b), (c), and (d) constant determine:
 - (1) Horsepower of steam cylinder, if steam driven.....i.hp.
 - (2) Horsepower of motor, if motor driven.....b.hp.
 - (3) Horsepower of air cylinders.....i.hp.
 - (4) Absolute pressure at exhaust steam nozzle.....lb. per sq. in.
 - (5) Absolute pressure at discharge air nozzle.....lb. per sq. in.
 - (6) Absolute pressure at air suction nozzles, with closed air suction and handling various quantities of free air at 60°F. and 14.7 lb. per sq. in. abs. as measured by orifices or gasometer..lb. per sq. in.
- (f) Plot curves:
 - (1) Air handled *vs.*¹ absolute pressure at suction nozzle.....
 - (2) Air handled *vs.* i.hp. of steam cylinders.....
 - (3) Air handled *vs.* b.hp. if motor driven.....
 - (4) Air handled *vs.* mechanical efficiency.....
 - (5) Air handled *vs.* volumetric efficiency.....

2. Hydraulic Types:

- (a) Absolute steam pressure at throttle.....lb. per sq. in.
- (b) Quality steam at throttle °F. superheat or per cent moisture.....
- (c) Speed (at which pump is designed to operate).....r.p.m.
- (d) Temperature of hurling water.....°F.
- (e) Quantity of hurling water.....gal. per min.
- (f) With items (a), (b), (c), (d), and (e) constant determine:
 - (1) Horsepower of driver (turbine or motor).....b.hp.
 - (2) Absolute pressure of hurling water at suction nozzle.....lb. per sq. in.
 - (3) Absolute pressure at pump discharge nozzle.....lb. per sq. in.
 - (4) Absolute pressure at air suction nozzle, with closed air suction and handling various quantities of free air at 60°F. and 14.7 lb. per sq. in. abs. as measured by orifice or gasometer.....lb. per sq. in.
- (g) Plot curves:
 - (1) Air handled *vs.*¹ absolute pressure at suction nozzle.....
 - (2) Air handled *vs.* b.hp. of driver.....
- (h) Data for other curves can be determined, if desired, by varying amount of hurling water, or temperature of hurling water.....

3. Steam Air Ejectors:

- (a) Absolute steam pressure at throttle.....lb. per sq. in.
- (b) Quality of steam at throttle °F. superheat or per cent moisture
- (c) Quantity of steam used by first-stage nozzles.....lb. per hr.
- (d) Quantity of steam used by second-stage nozzles.....lb. per hr.
- (e) Temperature of water entering intercooler (if any, surface or jet).....°F.

¹ The use of *vs.* (*versus*) in connection with curve plotting means that the two varying quantities mentioned, one preceding it and the other following, are to be used as the coordinates for the curve to be plotted.

- (f) Quantity of water used in intercooler.....gal. per min.
- (g) Amount of condensate from intercooler [item (c)] (if surface).....lb. per hr.
- (h) With items (a), (b), (c), (d), (e), (f), and (g) constant determine:
 - (1) Absolute pressure at main air suction nozzles.....lb. per sq. in.
 - (2) Absolute pressure at second-stage suction nozzles.....lb. per sq. in.
 - (3) Temperature of air vapor at second-stage suction nozzles.....°F.
 - (4) Absolute pressure at exhaust nozzles.....lb. per sq. in.
 - (5) Temperature of water leaving intercooler.....°F.
 - (6) Temperature of condensate leaving intercooler, with closed air suction and handling various quantities of free air as measured by orifice or gasometer.....°F.
- (i) Plot curves:
 - (1) Air handled *vs.*¹ absolute pressure at main suction nozzle.....
 - (2) Air handled *vs.* absolute pressure at second-stage suction nozzle.....
- (j) Data for other curves can be determined, if desired, by varying the steam pressure at throttle, exhaust pressure or temperatures and quantity of water used by intercooler.....
- (k) NOTE.—To conduct this test it is necessary to have a receptacle under vacuum into which the drains from the intercooler can be discharged. Air pumps operating in connection with condensers do not handle dry air but saturated air so that an independent or shop test handling dry or free air is not a criterion of the effectiveness of the apparatus in actual operation. These tests are made generally to determine if the maker's guarantee has been fulfilled, as guarantees on the air-removal apparatus are usually based on free-air handling capacity. Independent or shop tests can be made with air saturated by mechanical means but it is a very difficult matter to duplicate condenser conditions and a code covering this method of testing air-vapor removal apparatus has not been included.

A.S.M.E. TEST CODE (1923) FOR FEED-WATER HEATERS (ABRIDGED)

This code applies to open and closed boiler feed-water heaters, and with slight modifications to suit special conditions it may apply to heaters for heating water for any purposes when the heating element is either live or exhaust steam. In the open-type heater the heating steam mixes directly with the water to be heated. In the usual arrangement of closed-type heater the water passes through tubes surrounded by the heating system, though this arrangement may be reversed.

A feed-water heater is designed to heat a certain quantity of water through a given temperature range with a certain steam temperature or pressure available, with a limited loss of friction head in water flowing through the heater, and with a limited loss of steam through vents or stack.

There may be several objects of the tests, but usually they are conducted: (a) to determine whether the heater meets the designed conditions; (b) to determine the variation of temperature rise of water and friction drop in water with capacity in the case of closed heaters; (c) to determine the closeness with which the outlet-water

¹The use of *vs.* (*versus*) in connection with curve plotting means that the two varying quantities mentioned, one preceding it and the other following, are to be used as the coordinates for the curve to be plotted.

temperature approaches the steam temperature corresponding to the pressure in the open heater.

The heat-transfer coefficient is a useful and interesting item in connection with the analysis of a test of a closed feed-water heater. The heat-transfer coefficient is not, however, the sole measure of the merit of a heater.

Instruments and apparatus required for a heater test consist of the following:

- a. Barometer, preferably of the mercurial type.
- b. Mercury columns for measuring vacuums and low pressures, having scale graduations of not greater than 0.1 inch with vernier attachment.
- c. Bourdon gages for measuring pressures when too high for mercury columns.
- d. Thermometers: (1) For determining temperatures of feed water, condensate and vapors—graduated by half degrees and with scale readings from 32 to 300 or 350°F. (2) For determining steam temperature, graduated in degrees with scale readings from 32 to 300 or 350°F.
- e. Tanks and platform scales for measuring water (or water meters calibrated in place under conditions of use).
- f. Steam calorimeter, throttling or separating, depending upon amount of moisture present and pressure.
- g. Apparatus for testing oxygen content of water.¹

It is desirable that, wherever possible, observation of rate of water flow, pressures, and temperatures be obtained by continuous recording instruments, in addition to the observations made by the more precise instruments for instantaneous measurements.

Precisely the same testing apparatus for measuring the quantity of steam condensed in the case of closed heaters may be employed as that used for testing the engine, turbine, or other steam machinery supplying the feed-water heater (see Codes for Steam Engines, Steam Turbines, etc.). For measuring feed water the same kind of apparatus may be employed as that used in pumping-engine or water-wheel tests, such as weirs, nozzles, orifices, and Pitot tubes (page 184).

Open Heaters—Installation, Test, and Calculation of Results.—

In the open heater, which is also sometimes referred to as the "direct-contact" heater, in which the steam mixes directly with the water, it is possible so to design the water-distributing system that, over a wide range of capacity, the water will be heated to within a few degrees of the steam temperature corresponding to the pressure in the heater. In most open feed-water heaters the purpose is not only to heat the water but to filter or treat the water as well, and in these cases arrangements must be made to obtain analyses of the water entering and leaving the heater. Open feed-water heaters are also frequently used for partial deaeration of the water, and in such cases means should be provided for sampling the water and determining its oxygen content.¹

Inasmuch as the open feed-water heater serves as a storage tank supplying the boiler-feed pumps, the time required to empty the heater during normal operating conditions in case of failure of the supply of water to the heater is very important in establishing the rated capacity and size of heater. Of equal importance in establishing heater capacity is the time lag between the occurrence of an insufficiency of steam for a

¹ The determination of the oxygen content of feed water is a chemical test. It is not difficult and may be made by an unskilled operator.

desired outlet temperature and the appearance of this improper temperature at the water outlet.

Closed Heater—Installation, Test, and Calculation of Results.—

In a closed feed-water heater there is a definite relation between the capacity and outlet-water temperature which is determined by the ability of the surface to transfer heat. This ability of the surface to transfer heat under any given set of conditions is measured by the heat-transfer coefficient. The principal items affecting the value of the heat-transfer coefficient are (a) tube type, length, and arrangement; (b) water velocity; (c) condition of tubes as regards the presence of a film of scale, oil, or dirt; and (d) the presence of accumulated air in the steam space of the heater.

During the test of the closed heater, as well as during operation, provision should be made to keep the shell thoroughly drained of condensate and vented of air.

The thermometer in the steam space must not be installed where there is liable to be an air pocket nor may it be near a cold-water manifold. If near cold tubes it must be shielded for radiation.

Calculation of Results.—The logarithmic heat-transfer coefficient expressed in B.t.u. per hour per square foot of surface per degree of *logarithmic mean temperature difference* (page 492) is computed from the following formula:

$$K = \frac{w}{S} \log_e \frac{t_s - t_i}{t_i - t_o}$$

in which K = heat transfer coefficient.

w = pounds of water per hour.

S = heating surface, measured on the outside of the tubes, in square feet.

t_s = steam temperature in heater (if superheated steam is supplied, use temperature of saturated steam at the pressure in heater).

t_i = inlet water temperature.

t_o = outlet water temperature.

After working up the items tabulated, it is desirable to plot the results on *logarithmic paper* (page 528). The heat-transfer coefficient when plotted against the velocity of water in the tubes almost invariably gives a straight line on logarithmic paper. This curve then gives the data for determining the constant a and the exponent n in the equation.

$$K = av^n,$$

in which K = heat-transfer coefficient and v = velocity of water, in feet per second.

Friction drop when plotted against velocity usually gives a straight line on logarithmic paper which serves to determine the constant b and the exponent m in the friction drop formula,

$$H = bLv^m,$$

in which H = total friction drop, in pounds per square inch.

L = length of a single tube, in feet (if multipass, *length of total path of travel of water in heater*).

v = velocity of water, in feet per second.

TABLE XXXIX.—DATA AND RESULTS OF TEST CODE FOR OPEN FEED-WATER HEATERS

General Information

(1) Date of test; (2) location of plant; (3) owner; (4) builder; (5) test conducted by, (6) object of test.

Descriptions and Dimensions, Data, Etc.

7. Type of heater.....	
8. External dimensions of heater.....	
9. Gross volume of heater.....	cu. ft.
10. Weight of heater, empty.....	lb.
11. Weight of heater when operating (including water).....	lb.
12. Volume of steam space.....	cu. ft.
13. Shape and dimensions of steam inlet opening into heater.....	in.
14. Size of water inlet.....	in.
15. Size of water outlet.....	in.
16. Material of shell.....	
17. Volume of water in heater at operating water level.....	cu. ft.
18. Volume of water between overflow level and level at which make-up valve will open.....	cu. ft.
19. Rated capacity of heater, water per hour.....	lb.
20. Location of thermometers.....	
21. Number and arrangements of baffles or trays.....	
22. Description of filtering or purifying arrangement.....	
23. Size and arrangement of venting connections.....	
24. Description of metering apparatus installed in heater.....	
25. Description of automatic steam and water control.....	
26. Description of water-distributing boxes.....	
27. Nature and amount of insulation on heater.....	
28. Duration of test.....	
29. Barometer reading as observed.....	in. of mercury
30. Room temperature.....	°F.
31. Quantity of water admitted to heater.....	lb. per hr.
32. Inlet-water temperature.....	°F.
33. Outlet-water temperature.....	°F.
34. Steam pressure in heater, gage.....	lb. per sq. in. or in. of mercury
34a. Absolute steam pressure in heater.....	lb. per sq. in.
35. Steam temperature in heater, by thermometer.....	°F.
36. Steam temperature at inlet, by thermometer.....	°F.
37. Steam used per hour.....	lb.
38. Total water discharged from heater.....	lb. per hr.
39. Volume of water in mixing compartment of heater.....	cu. ft.
40. Volume of water in storage compartment of heater.....	cu. ft.
41. Pressure drop from steam end of heater to vent end.....	lb. per sq. in.
42. Time lag between occurrence of steam deficiency and change of outlet temperature.....	
43. Analysis of water entering heater.....	
44. Analysis of water leaving heater.....	
45. Oxygen content of water entering heater.....	
46. Oxygen content of water leaving heater.....	

Computed and Deduced Results

47. Steam temperature corresponding to absolute pressure in heater.....	°F.
48. Quality of steam supplied to heater per cent moisture or °F. superheat	
49. Temperature difference between steam temperature corresponding to heater pressure and outlet water temperature.....	°F.
50. Steam theoretically required per pound water.....	lb.
51. Steam used per pound water, actual.....	lb.
52. Time required to empty heater when operating at rated capacity.....	hr.
53. Steam lost up stack.....	lb. per hr.

TABLE XL.—DATA AND RESULTS OF TEST CODE FOR CLOSED FEED-WATER HEATERS

General Information

(Same as for Table XXXIX)

Descriptions and Dimensions, Data, Etc.

7. Type of heater.....	
8. Position of heater, horizontal or vertical.....	
9. Condition of heating surface.....	
10. Number of tubes.....	
11. Number of passes.....	
12. Length of single tube.....	ft., in.
13. Distance of travel of water through heater.....	ft., in.
14. Special type of tube, description.....	
15. Outside diameter of tube.....	in.
16. Thickness of tube.....	in.
17. Heating surface, of tubes, outside of tubes.....	sq. ft.
18. Diameter of heater over shell.....	ft., in.
19. Length of heater over shell.....	ft., in.
20. Thickness of shell.....	in.
21. Material of tubes.....	
22. Material of shell.....	
23. Weight of heater, empty.....	lb.
24. Weight of water in heater.....	lb.
25. Gross volume of heater.....	cu. ft.
26. Shape and dimensions of steam inlet opening into heater.....	in.
27. Arrangement of steam supply pipes into heater.....	
28. Size of water inlet and outlet.....	in.
29. Size of drain.....	in.
30. Type and size of drain trap.....	
31. Location and type of air vents.....	
32. Arrangement of baffles.....	
33. Nature and amount of insulation on heater.....	
34. Location of thermometers.....	
35. Duration of test.....	
36. Barometer reading as observed.....	in. of mercury
37. Room temperature.....	°F.
38. Quantity of water through heater.....	lb. per hr.
39. Steam pressure in heater, gage.....	lb. per sq. in. or in. of mercury
39a. Absolute steam pressure in heater.....	lb. per sq. in.
40. Steam temperature in heater by thermometer.....	°F.
41. Drain temperature.....	°F.
42. Vent temperature by thermometer.....	°F.

- 43. Steam temperature at inlet by thermometer °F.
- 44. Inlet-water temperature °F.
- 45. Outlet-water temperature °F.
- 46. Weight of steam used lb. per hr.
- 47. Inlet-water pressure, gage lb. per sq. in.
- 48. Outlet-water pressure, gage lb. per sq. in.
- 49. Water-pressure drop by differential mercury column in.

Computed and Deduced Results

- 50. Velocity of water in tubes ft. per sec.
- 51. Steam temperature in heater corresponding to absolute steam pressure °F.
- 52. Quality of steam supplied to heater ... per cent moisture or ... °F of superheat
- 53. Temperature rise °F.
- 54. Logarithmic heat-transfer coefficient, in B.t.u. per hour per square foot of surface,
per degree of logarithmic mean temperature difference (p. 503)
- 55. Weight of steam theoretically required per pound of water, computed from heat
balance lb.
- 56. Weight of steam used per pound of water lb.
- 57. Water-pressure drop in heater lb. per sq. in.
- 58. Steam lost up stack lb. per hr.

CHAPTER XX

TESTING OF ATMOSPHERIC WATER-COOLING EQUIPMENT— COOLING TOWERS, SPRAY NOZZLE INSTALLATIONS AND COOLING PONDS (ABRIDGED)

This Code is designed to be used when conducting tests of atmospheric water-cooling equipment in which heat transfer from water to the atmosphere occurs when the fluids are in contact, and is mainly due to evaporation.

There is a large variety of apparatus employed for this purpose, of which the following are examples: (a) chimney or natural-draft cooling towers; (b) mechanical-draft cooling towers, with or without chimney; (c) atmospheric or wind towers, with or without chimney; (d) spray-nozzle installations; (e) natural or artificial ponds and reservoirs, with or without spray nozzles.

When the object is to obtain data to be used for the calculation of water-cooling effect available for power or for other purposes, investigations covering the cooling effected by surface evaporation in ponds, lakes, or reservoirs under various atmospheric conditions may be complicated where there is an auxiliary cooling apparatus such as a spray system installed over a pond or reservoir.

Water-cooling equipment is usually guaranteed to cool a definite quantity of water through a definite temperature range with definite temperatures of air and humidities (or wet-bulb temperatures), and with some types of apparatus it is sometimes necessary to specify wind velocities. A guarantee is also frequently made covering the amount of make-up water required to allow for losses and the power consumed by auxiliaries.

Ordinarily the total water loss is made up of two components: (1) evaporative loss due to evaporation of a definite fraction of the water cooled in giving up its heat to the air and (2) windage loss or loss due to the mechanical entrainment of water globules in the air current.

The evaporative loss is a function of the cooling range and the wet-bulb temperature, so that, where the cooling plant serves condensing apparatus, this loss varies from 60 per cent (in winter) to 90 per cent (in summer) of the condensate. The evaporative loss may be calculated and subtracted from the total loss to determine windage and other losses.

As definite guarantee atmospheric conditions are practically impossible to duplicate in a test, it is desirable that curves be furnished to show the guaranteed performance of the apparatus over a considerable range

of atmospheric conditions, when varying quantities of water are cooled. To accomplish this, reference to the **wet-bulb temperature** of the atmosphere has practically superseded reference to the dry-bulb temperature and relative humidity as a basis for stating cooling-tower performance, for all types of cooling equipment *except* the chimney or natural-draft cooling tower.

For chimney or natural-draft cooling towers, however, it is not strictly permissible to refer performance to wet-bulb temperatures alone on account of the variation in density of the atmosphere at any given wet-bulb temperature with different combinations of dry-bulb temperatures and relative humidities and the effects upon the draft of this variation of atmospheric density. For such towers, therefore, it will be necessary to refer performance to a *combination* of the dry-bulb or the wet-bulb temperature with the relative humidity as a basis.

To obtain wet- and dry-bulb air temperatures for determining humidity, a sling psychrometer (page 441) or other approved mechanically ventilated dry- and wet-bulb thermometers should be used. All readings should be taken to windward of the cooling plant, thus avoiding the warm moist air discharged by it, which raises the wet-bulb temperature to leeward and so might introduce a serious error in an observation taken in that direction.

It is important that the muslin covering for the wet-bulb thermometer be kept in good condition. The evaporation of water from the muslin always leaves in its meshes a small quantity of solid material which sooner or later somewhat stiffens the muslin so that it does not readily take up water. This will be the case if the muslin does not readily become wet after being dipped in water. On this account it is desirable to use as pure water as possible—preferably distilled water—and also to renew the muslin from time to time. New muslin should always be washed to remove sizing before being used. If desired, knitted wet-bulb jackets can be obtained from thermometer dealers.

To make an observation, the wet bulb is thoroughly saturated with water by dipping it into a small cup or wide-mouth bottle. The thermometers are then whirled rapidly for 15 or 20 seconds, stopped, and quickly read, the wet bulb first. This reading is kept in mind, the psychrometer immediately whirled again, and a second reading taken. This is repeated three or four times, or more, if necessary, until at least two successive readings of the wet bulb are found to agree very closely, thereby showing that it has reached its lowest temperature. A minute or more is generally required to secure the correct temperature.

The sling psychrometer will give an accurate reading of the true wet-bulb or equilibrium temperature of water evaporating adiabatically in air, if swung at a rotational velocity at the bulb of not less than 1,000 feet per minute.¹

¹ See paper by W. F. Carrier and Daniel C. Lindsay, Temperatures of Evaporation of Water into Air, *Trans. A.S.M.E.*, 1924.

When the air temperature is near the freezing point, it often happens that the temperature of the wet-bulb thermometer will fall several degrees below the freezing point, but the water will still remain in a liquid state. No error results from this, provided the minimum temperature is reached. If, however, as frequently happens, the water suddenly freezes, a large amount of heat is liberated, and the temperature of the wet bulb immediately becomes 32°F. In such cases it is necessary to continue the whirling until the ice-covered bulb has reached a minimum temperature.

If a psychrometer is used out-of-doors *when the sun is shining*, it should be whirled in the shade of a building or tree or, as may sometimes be necessary, under an umbrella. In all cases, there should be perfectly free circulation of the air, and the observer should face the wind, whirling the psychrometer in front of his body. It is a good plan, while whirling, to step back and forth a few steps to further prevent the presence of the observer's body from giving rise to erroneous observations.

The determination of wet- and dry-bulb temperatures at the discharge of cooling towers is extremely difficult on account of particles of water which are mechanically entrained in the air and also because of eddy currents which are frequently induced by the comparatively large discharge openings. The thermometers used at the discharge of a cooling tower should be protected by some kind of moisture separators to remove the mechanically entrained water and also to shield them from extraneous wind or eddy currents. Results of calculations based on wet- and dry-bulb temperatures obtained at the discharge of a cooling tower, unless unusual precautions are taken, must be used with caution.

Air *velocities* and *quantities* may be measured by means of an *anemometer* by dividing a cross section of the air current into comparatively small areas and determining the air velocity in each small area. The velocities thus obtained are then averaged to determine the average velocity over the entire cross section. This method of determining air quantities is subject to many difficulties in its operation, and the results obtained, unless great care is taken to guard against eddy currents, should be considered as approximate only.

In testing wind towers and spray-nozzle and surface-cooling plants, select a point to windward of the plant having unobstructed exposure to the breeze and make 5-minute *anemometer* runs.

If a test of a spray installation is made, the spray heads should be thoroughly cleaned before proceeding with the test. Approximate determination of the nonclogging characteristics of a spray nozzle may be obtained by passing steel balls of various sizes through the water channel. For practical operation it may be assumed that the nozzle which will pass the larger ball freely through its water passage will be least subject to clogging by foreign material.

Calculation of Results.—The calculation of heat balance is only possible for that apparatus with which it is possible to obtain compara-

tively accurate data covering the air condition leaving the tower (see page 509) and does not apply to wind towers, spray-nozzle plants, or surface-cooling ponds. Since the heat given up by the water must equal that absorbed by the air (radiation being considered negligible), the heat balance as calculated from the data obtained must show a practical agreement between these quantities. Also the amount of air required (Item 76 in the tabulation below) as estimated from the *heat balance* should agree with the amount of air entering the tower as measured (Items 64 and 77).

Let W = pounds of water entering cooling tower per minute calculated from Item 44 in the A.S.M.E. Test Code (page 512).

A = pounds of dry air entering cooling tower per minute (calculated)¹ from Items 36, 37, 38, 39 and 64.

t_{w1} = temperature of water entering tower (Item 45).

t_{w2} = temperature of water leaving tower (Item 46).

t_{a1} = temperature of air entering tower (Item 37).

t_{a2} = temperature of air leaving tower (Item 66).

S = specific heat of air (0.242 B.t.u. per pound).

M_1 = specific humidity at entrance (pounds of vapor per pound of air),¹ calculated from Items 36, 37, 38, 39, and psychrometric chart (page 442).

M_2 = specific humidity at exit (pounds of vapor per pound of air), calculated from Items 36, 66, 67, 68, and psychrometric chart.

H_{m1} = heat content (enthalpy)² of M_1 pounds of vapor at temperature t_{a1} ; calculated from Items 36, 37, 38, 39, steam table (Appendix), and psychrometric chart.

H_{m2} = heat content (enthalpy) of M_2 pounds of vapor at temperature t_{a2} ; calculated from Items 36, 66, 67, 68, steam table and psychrometric chart.

H_{w1} = heat content (enthalpy) of cooling water entering tower per pound of dry air entering tower.

H_{w2} = heat content (enthalpy) of cooling water leaving tower per pound of dry air entering tower.

H_{a1} = heat content (enthalpy) of moist air per pound of dry air entering tower.

H_{a2} = heat content (enthalpy) of moist air per pound of dry air leaving tower.

¹ For similar calculations requiring the determination and the use of the relative humidity of air, see Moyer and Fittz, "Air Conditioning," McGraw-Hill Book Company, Inc., New York, 1933.

² All values of heat content (enthalpy) are referred to dry air at 32°F. or to saturated liquid water at 32°F. For definitions of **Enthalpy** quantities, see Moyer, Calderwood and Potter, "Elements of Engineering Thermodynamics," 5th ed., p. 74, John Wiley & Sons, Inc., New York, 1932).

Then $H_{w1} - H_{w2} = H_{a2} - H_{a1}$:

$$H_{w1} = \frac{W}{A}(t_{w1} - 32) \text{ and } H_{w2} = \frac{W}{A}(t_{w2} - 32) - (M_2 - M_1)(t_{w2} - 32).$$

$$H_{w1} - H_{w2} = \frac{W}{A}(t_{w1} - t_{w2}) + (M_2 - M_1)(t_{w2} - 32).$$

$$H_{a1} = (t_{a1} - 32)S + H_{m1}.$$

$$H_{a2} = (t_{a2} - 32)S + H_{m2}.$$

$$H_{a2} - H_{a1} = S(t_{a2} - t_{a1}) + (H_{m2} - H_{m1}).$$

$$\frac{W}{A}(t_{w1} - t_{w2}) + (M_2 - M_1)(t_{w2} - 32) = S(t_{a2} - t_{a1}) + (H_{m2} - H_{m1}).$$

$$A = \frac{W(t_{w1} - t_{w2})}{S(t_{a2} - t_{a1}) + (H_{m2} - H_{m1}) - (M_2 - M_1)(t_{w2} - 32)}.$$

A.S.M.E. TEST CODE (1929) FOR ATMOSPHERIC WATER-COOLING EQUIPMENT

General Information

(1) Date of report; (2) place of test, prevailing direction and mean velocity, in miles per hour, of summer winds at site; (3) cooling water for; (4) object of test; (5) test conducted by; (6) name of manufacturer.

Description and Dimensions

Cooling Tower: (7) Compass bearing of major axis on all wind towers; (8) type of tower; (9) dimensions horizontal and vertical sections including chimney, if any; (10) size and depth of reservoir (if any); (11) type of water distributor; (12) location of water distributor; (13) type of filling or splash and distributing decks; (14) arrangement and depth of filling; or number, dimensions and spacing of splash, and distributing decks; (15) material of filling or decks; (16) distance from center of distributor to water level in reservoir; (17) number and size of fans (if any); (18) type of fans, dimensions and drive; (19) location of fans.

Surface or Spray Cooling: (20) Compass bearing of major axis on all plants; (21) type of spray nozzles, if any; (22) number and size of spray nozzles; (23) number of nozzles and arrangement in group; (24) distance between spray groups and group lines; (25) height of spray nozzles above water level in reservoir; (26) size of reservoir (at water line), in feet; (27) area of reservoir, in square feet; (28) average depth of reservoir, in feet; (29) capacity of reservoir, in cubic feet; (30) baffle arrangement in reservoir (if any); (31) location of inlet and outlet from reservoir; (32) source of make-up water; (33) special features.

TABLE XLI.—DATA AND RESULTS OF WATER-COOLING EQUIPMENT TESTS

Data and Duration

(34) Date of test; (35) duration of test in hours

Average Atmospheric Conditions

(36) Barometer, in inches of mercury (corrected for temperature); (37) temperature of atmosphere, in degrees Fahrenheit; (38) temperature of wet-bulb thermometer, in degrees Fahrenheit; (39) relative humidity, in per cent; (40) direction of wind; (41) velocity of wind, in miles per hour; (42) weather (clear or cloudy); (43) rainfall during test, in inches (if any).

All Equipment Test Data

44. Amount of water cooled.....gal. per min.
45. Temperature of water entering sprays, tower, or reservoir.....°F.
46. Temperature of water leaving reservoir.....°F.
47. Make-up water used.....gal. per min.
48. Temperature of make-up water.....°F.
49. Level of water in reservoir at start of test.....ft.
50. Level of water in reservoir at end of test.....ft.
51. Temperature¹ of water in reservoir at start of test.....°F.
52. Temperature¹ of water in reservoir at end of test.....°F.
53. Loss of water in per cent of total circulated.....per cent
54. Overflow from reservoir (if any).....gal. per min.
55. Area drained by reservoir.....acres
56. Average run-off.....per cent of rainfall
57. Reservoir leakage.....
58. Pressure of water at center line of distributor or at the spray nozzles.lb. per sq. in.
59. Pressure of water at entrance of tower or spray system.....lb. per sq. in.
60. Total water head (p. 548) required for operation.....ft.

Mechanical Draft and Chimney Towers

61. Speed of fans (if any).....revolutions per min.
62. Power required to drive fans.....hp.
63. Steam required to drive fans.....lb. per hr.
64. Air handled.....cu. ft. per min.
65. Air pressure or draft in base of tower.....in. water
66. Dry-bulb temperature of air leaving tower.....°F.
67. Wet-bulb temperature of air leaving tower.....°F.
68. Relative humidity of air leaving tower.....per cent
69. Water cooled per square foot of cross section of tower, horizontal.gal. per min.
70. Air per square foot of cross section of tower.....cu. ft. per min.

Heat Balance

71. Water entering tower per pound of dry air per minute (calculated from Items 44, 64, 37, and 39).....lb.
72. Vapor per pound of dry air entering tower (calculated from Items 36, 37, 38, 39, and psychrometric chart (p. 442)).....lb.
73. Vapor per pound of dry air entering tower (calculated from Items 36, 66, 67, 68, and psychrometric chart).....lb.
74. Heat given up per pound of water entering tower.....B.t.u.
75. Heat absorbed per pound of dry air entering tower.....B.t.u.
76. Air at temperature (Item 37) and humidity (Item 39) required as calculated from heat-balance equation.....cu. ft. per min.
77. Air at temperature (Item 37) and humidity (Item 39) as determined from actual air measurement.....cu. ft. per min.

¹ It is advisable to measure the temperature of water in a reservoir or pond at different places and depths to determine if possible the path of circulation and how much of the volume of the pool is used effectively.

CHAPTER XXI

TESTS OF HOISTS, BELTS, AND FRICTION WHEELS

Hand-operated Hoists.—The simplest form of hand hoist is the block and tackle, consisting of two “blocks” (upper and lower), which serve as frames for mounting one or more pulleys on each. The upper or stationary block is provided at the top with a hook for attaching the block and tackle to an overhead support, while the lower or movable block has at the bottom another hook for the attachment of the load to be raised. One end of the hoisting rope is fastened to the upper block, the other end being wound alternately over the pulleys in the upper and lower frames until it hangs free. The lifting force is applied to this free end.

Chain Hoists.—The block-and-tackle hoist will run backwards when the lifting force on the free-end is removed. To avoid this disadvantage, chain hoists have been designed that have automatic load-supporting power. A hoist of this kind has an upper hook for attachment to an overhead support, a lower hook for the attachment of the load, and an endless chain for supporting the load and the application of the lifting force, as shown in Figs. 293 to 294. The one shown in Fig. 293 is a *differential hoist*, which has in the upper block two metal sheaves that are rigidly joined. These sheaves are of slightly different diameters and each has *link pockets* around its circumference into which the links of the chain fit to prevent the slipping of the chain on these sheaves. The lower block has only one sheave which is merely grooved like those in a block and tackle (not notched with link pockets), thus allowing the chain to *slide* so that the lower block may always be at the bottom of the loop of the chain supporting it.

The automatic load-supporting power of this hoist is due to the small difference in diameter of the sheaves in the upper block, the larger one having one extra link pocket. This makes the multiplication ratio of the power applied to the load carried too small to overbalance the friction of the parts, so that the load is automatically sustained at any point, making the hoist self-locking. In the differential hoist there is *one* continuous chain.

The **worm hoist** (sometimes called “duplex”) consists of a link-pocket sheave *B* mounted on the shaft of a *worm* (Fig. 293a). A continuous



FIG. 293.
Differential
hoist.

chain *A* passes over the sheave *B*. The lifting power is transferred from this chain, first, to the worm and then to two link-pocket sheaves attached to opposite ends of the shaft of the *worm wheel*. It has *two* continuous chains, one for the sheave *B* (lifting power) and the other for the two load sheaves *D*. The load chain is attached to the swivel

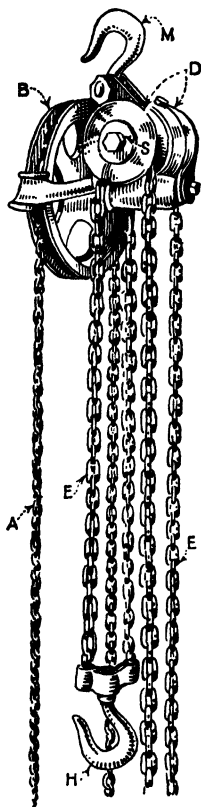


FIG. 293a.—Duplex hoist.

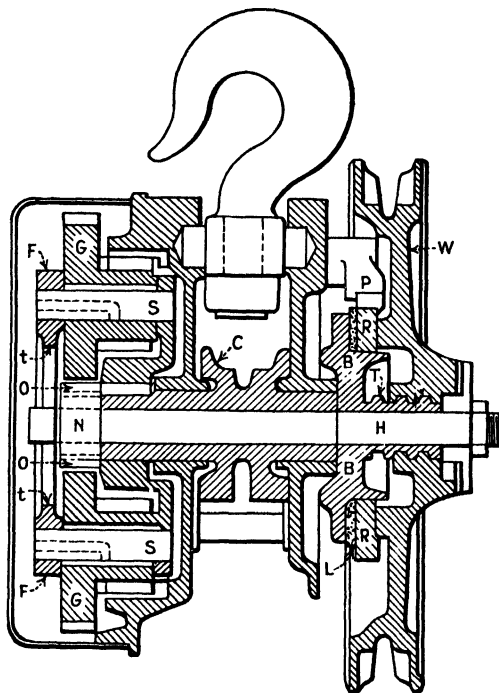


FIG. 294.—Upper block of triplex hoist.

hook *H*. Its two ends are fastened to the hook for supporting the load as shown in the figure and pass over the two load sheaves *D*, with the slack hanging in a loop. The load is raised or lowered by the lifting force applied to opposite sides of this loop. The non-reversability of worm gears is utilized to sustain the load in any position.

The **spur-gearred hoist** (sometimes called "triplex") is shown in Fig. 294. In this hoist the lifting force is transmitted through a spur-gear train, "sun-and-planet" gears,¹ and load-lifting sheaves. Two separate chains are used, one for the attachment of the load and another

¹ For explanation of "sun-and-planet" gears in similar devices, see James A. Moyer, "Gasoline Automobiles," 4th ed., p. 390, McGraw-Hill Book Company, Inc., New York, 1932.

for the lifting force. The load is held in any position by the operation of a mechanical brake, consisting of a ratchet and pawl that engage on friction plates. To lower a load, the lifting or operating chain is pulled in the reverse direction from that for raising the load, thus disconnecting the friction plates. It has a higher efficiency than other types.

In the operation of this hoist the lifting force is applied by the operating chain to its sheave and then to the shaft H , to which the lifting sheave is attached. This lifting force is then transmitted to the two spur gears G, G on the short shafts S, S attached to the circular frame F keyed to the hub of the hoisting chain collar C . Attached to the face of each of the gears G, G are small intermediate gears O, O (shown by dotted lines). These intermediate gears serve to rotate the circular gear frame F , by rolling around the face of a gear with internal teeth (t, t). This gear is a part of the main frame of the hoist. This combination of gears makes the sun-and-planet system. The automatic locking device in this hoist is designed to operate by the force of gravity of the load, and consequently there is less friction in its operation than in other types. This braking mechanism operates so that, when the load is raised, the hand wheel W advances on the screw thread T of the brake collar B , thus clamping the leather disk L to the ratchet R , which operates as a unit with the brake collar B and the hand wheel W when a load is being raised.

The brake collar B is attached to the chain-wheel shaft H . When a load has been raised as far as required, the load is prevented from falling by the engagement of the pawl P on the ratchet wheel R ; and as this ratchet wheel is clamped to the brake collar B , the load is automatically prevented from falling, until a backward pull is exerted on the hand chain to relieve the pressure on the pawl P .

Velocity Ratio of Hoists.—Differential hoists (see Fig. 295) are more complicated than the ordinary chain or rope hoist. In this apparatus, as shown in the standard books on the theory of mechanism, the velocity ratio V_r is expressed according to the dimensions in the figure by

$$V_r = \frac{2R_1}{R_1 - R_2}.$$

It is difficult, however, to measure accurately the radius of these wheels on account of the irregular surface made for gripping the links. Now since the *circumferences* of these wheels are proportional to the *radii*, the *velocity ratio* V_r may be determined by counting the number of link pockets in each of the wheels, and its value will be given as shown by the equation by the ratio of twice the number of link pockets in the larger

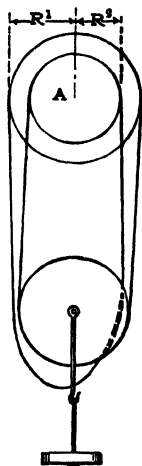


FIG. 295.—Differential hoist.

wheel divided by the difference between the number of link pockets in the larger and the smaller wheels.

In other types of hoists where the foregoing method is not applicable and the diameters cannot be readily measured, the velocity ratio can be determined by tying a piece of string on a link of the "load" chain (or rope), as the case may be, opposite some fixed part of the hoist and marking in the same way a point on the chain (or rope) to which the pull is applied. Now when the "load" chain has been moved a measured distance, the corresponding movement of the lifting force can be measured. Several observations should be made to eliminate probable errors in the measurements. Velocity ratio is usually expressed by making the second member of the ratio unity, thus 13 to 1, 4 to 1, etc.

Efficiency of Hoists.—An efficiency test of a hoist is made by determining the ratio between the work done in lifting the load to that applied to the hand chain (or rope). Stated briefly, this determination is made by raising a known weight, observing at the same time, by means of a spring balance fastened to the hand chain (or rope), the pull or force required to keep the load moving after it has been started. Allowance must be made, of course, for the number of times the power is multiplied, that is, the relative velocity value.

On account of the great friction at starting, the reading of the spring balance should be made when constant after starting by hand.¹ Determinations should also be made when the load is being lowered, but these results should not be averaged with those for raising the load, because a hoist is generally used only for raising loads.

The actual or direct efficiency E of a hand hoist when *raising* a load is the ratio of the *actual* mechanical advantage M_a to the theoretical mechanical advantage M_t .

By using the following notation:

W = load on hoists, in pounds.

L = distance load is moved, in feet.

w = weight on hand chain to *raise* load, in pounds.

l = distance weight on hand chain is *raised*, in feet.

Actual mechanical advantage M_a of the hoist is

$$M_a = \frac{W}{w}.$$

Theoretical mechanical advantage M_t of the hoist is

$$M_t = \frac{l}{L}.$$

¹ If the spring balance is used in the inverted position, its weight must be added to the pull. Also the weight of the scale pans used for supporting the weights must be added to the load raised.

Actual efficiency E of hoist is

$$E = \frac{M_a}{M_t} = \frac{WL}{wl}$$

Friction work F of hoist is

$$F = wl - WL.$$

Determination of Tension in Belts and Rope Drives.—Tests are often required to determine the power transmitted by belts and ropes for specified conditions of load, speed, tension, and the coefficient of friction between these and the pulleys on which they run. Suitable apparatus for such tests consists of a device with which the belts or ropes can be operated with different tensions. Usually the load is not increased beyond the limit producing 3 per cent slip.¹ The initial tension in the belt or rope should be measured when at rest.

Testing of Belts.—For determining the qualities of belts as regards power transmission and the coefficient of friction, belt tests are desirable. Sometimes belts or pulleys of different materials are to be compared, while again tests may be required to determine the effect of belt dressings. The apparatus to be described is simple in construction and inexpensive. It consists essentially of two pulleys on the ends of a universal coupling. One of these pulleys is driven by a variable-speed motor to which it is belted. The other is set up on the frame shown in Fig. 296, suspended on a knife-edge bearing at O , so that the end at S is free to move in a vertical plane and any horizontal tension in the belt on the pulley which the apparatus supports produces a proportional pressure on a scales at S . The pulley shown in the figure is the driver for another pulley which is attached in fixed bearings to a shaft about 25 feet away. A Prony brake is supported on this shaft which is adjustable in its bearings, so that the tension in the belt can be varied.

Let T_1 = tension in tight side of belt, in pounds.

T_2 = tension in slack side of belt, in pounds.

W = pressure on scales at S , in pounds.

Then

$$(T_1 + T_2)a = Wb \quad \text{or} \quad T_1 + T_2 = \frac{Wb}{a}. \quad (100)$$

¹ Slip in belts or ropes is the ratio of the difference between the revolutions of the driver and the follower divided by the revolutions of the driver, each taken, of course, for the same time unit.

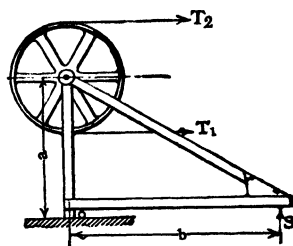


FIG. 296.—Belt and rope testing apparatus.

When power is being transmitted by the belt and absorbed by the Prony brake, we have, since the power transmitted is proportional to the difference of tension, by moments, $wR + T_2r = T_1r$, and then

$$T_1 - T_2 = \frac{wR}{r}, \quad (101)$$

where w = net weight on brake, in pounds.

R = length of brake arm, in feet.

r = radius of driven pulley, in feet.

Combining Eqs. (100) and (101),

$$T_1 = \frac{\frac{Wb}{a} + \frac{wR}{r}}{2}. \quad (102)$$

$$T_2 = \frac{\frac{Wb}{a} - \frac{wR}{r}}{2}. \quad (103)$$

These last equations give the **total** tensions T_1 and T_2 which are usually reduced to pounds per inch of width of the belt.

The **coefficient of friction** f between the belt and the pulley is given in books on mechanics as

$$f = \frac{\log \frac{T_1}{T_2}}{0.434 c}, \quad (104)$$

where c is the arc of contact of the belt on the pulley in inches divided by the radius of the pulley in inches.

Belts will be stretched more on the tight than on the slack side and this unequal stretching causes a difference in velocity or, per unit of time, a difference between the length of belt on the driving and driven pulleys. This difference is called the **creep** or **slip** of the belt. If r' is the radius and N' the revolutions per minute of the driving pulley, while r is the radius and N is the revolutions per minute of the driven pulley, then the difference in velocity is $2\pi r'N' - 2\pi rN$, because the length of belting coming on a pulley in a unit of time is equal to its peripheral speed. Slip or creep is expressed usually as a percentage of the speed of the driver so that

$$\text{Percentage slip or creep} = \frac{r'N' - rN}{r'N'}. \quad (105)$$

Efficiency of transmission = delivered horsepower $\div$ horsepower input.

Tests of Friction Wheels.—The apparatus frequently used for determining the **coefficient of friction** between friction wheels consists of a pair of pulleys, one of them at least usually made of some soft metal

like aluminum or a fibrous material like straw fiber, leather fiber, or paper. This driver runs on a follower, generally of some other material. Power delivered to the "follower" shaft is absorbed by a Prony brake. A bell-crank lever to which weights can be attached is used to hold or press the two pulleys together.

The coefficient of friction as determined by such an apparatus is the ratio of the tangential pull to the total normal pressure. If the coefficient of friction is represented by f , and other symbols are used as follows:

r_1 = the effective brake arm, in inches.

r_2 = the radius of driven pulley, in inches.

w = net weight on the brake, in pounds.

p = normal pressure, in pounds per inch of width of the narrower pulley.

z = width of the narrower pulley, in inches.

Then,

$$f = \frac{\frac{wr_1}{r_2}}{pz} = \frac{wr_1}{pr_2z}. \quad (106)$$

CHAPTER XXII

TESTING OF LUBRICANTS

In the space which can be taken for this chapter on the testing of oils, only a few of the simpler and more important tests can be given. To make a complete report on the composition of a sample of oil, it is generally advisable to submit the suspected sample to a competent professional chemist, preferably with a large and varied experience in the analysis of lubricants. Characteristic tests, however, will be given here which should be sufficiently serviceable to engineers to determine whether an oil is pure enough for the use intended and whether, if it is to be used as a lubricant, it has the properties essential for reducing friction to a minimum or, if used as a fuel, it is necessary to ascertain the degree to which it is susceptible to vaporization, and, in the case of very light oils, the temperature to which it may be exposed without danger of becoming inflammable. The **heating or calorific value of fuel oils** is also a very important test, but this has been explained in Chap. VIII. It remains therefore only to take up, for fuel oils, the tests for **specific gravity** and the **flash and burning points**.

Specific gravity of a liquid is determined most accurately in most cases where a sensitive chemist's balance is available by the use of a specific-gravity bottle. A conventional type is shown in Fig. 297. Bottles for this purpose are made with special care of thin glass and the weight of distilled water which they will contain is determined accurately and etched on the outside surface of the bottle. The bottle is provided with a small ground-glass stopper having a capillary tube or hole drilled through it, so that when the bottle is filled to the top of the capillary tube it will always hold the same volume of liquid.

In determining the specific gravity the bottle is filled with the liquid to be tested, care being taken to avoid the formation of air bubbles. The stopper is then inserted and some of the liquid will run out through the capillary tube. This excess should be wiped off so that the bottle will be clean and dry. It can then be weighed. After once determining the weight of the bottle filled with distilled water at 60°F. the bottle can be used without again weighing it with water.¹ The weight of the empty bottle should be ascertained from time to time to determine, more than for any other reason, whether it is clean. If it is found to

¹ This suggestion is made because it is not always easy for engineers to obtain clean distilled water. Condensed steam from a surface condenser is usually, however, sufficiently free from impurities and is, of course, distilled water.

weigh more than when new, obviously it needs cleaning. The weight of the liquid in the bottle when it is full, divided by the weight of the corresponding amount of distilled water is the specific gravity of the liquid being investigated. The liquid tested should be at 60°F. when it is weighed in the bottle, as this is the standard temperature for the specific gravities of all oils.

For the determination of the specific gravity of very thick oils and greases, a type of bottle or tube known as Hubbard's (Fig. 298) is often used. It consists of a metallic tube with a ground-in stopper, having a slightly larger bore than the capillary tube in the glass stopper of the ordinary specific-gravity bottle. In commercial practice the specific gravity of liquids is usually determined by means of an instrument called

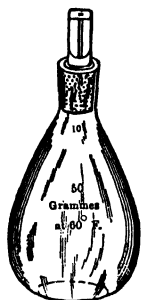


FIG. 297.—Specific-gravity bottle for thin oils.

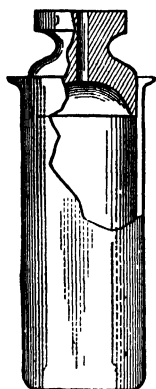


FIG. 298.—Specific-gravity bottle for thick oils.

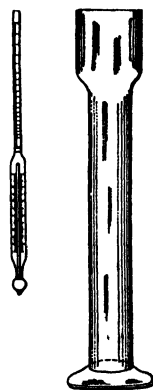


FIG. 299.—Hydrometer and hydrometer jar.

a **hydrometer** (Fig. 299). It is most conveniently used by filling a glass jar, preferably similar to the one in the figure, with the liquid to be tested and then inserting the hydrometer. The reading on the scale of this instrument, which is at the level of the surface of the liquid, is the specific gravity. The hydrometer shown has a thermometer combined with it, so that the temperature of the liquid can be read on the scale of the upper stem. The "surface of the liquid" is here understood to mean the surface of the main body of the liquid and not the level of the ring around the instrument due to capillarity. Hydrometers are made with two standard scales. One is the ordinary **specific gravity scale** graduated to correspond to the determinations of specific gravity as defined for determinations with the specific-gravity bottle; that is, it uses always the ratio of the weight of the liquid to the weight of an equal volume of water. The other is an arbitrary one known as **Baumé's** and is much used by trades people. For short, it is often called the "gravity" scale. The following condensed table will be found con-

venient for converting one scale to the other. As a "rule of thumb" key to the two scales, it will be found useful to remember that 70 degrees in the Baumé scale is approximately 0.70 in specific gravity.

TABLE XLII.—SPECIFIC GRAVITIES EQUIVALENT TO BAUMÉ DEGREES

Baumé degrees	Specific gravity	Baumé degrees	Specific gravity
10	1.0000	60	0.7368
20	0.9333	65	0.7179
30	0.8750	70	0.7000
40	0.8235	75	0.6829
50	0.7777	80	0.6666
55	0.7567	85	0.6511

In order that specific-gravity determinations may be accurately compared, the standard temperature of 60°F. has been adopted, which means that a statement of specific gravity of a substance at standard conditions implies that the value given of its specific gravity is that at the equivalent temperature of 60°F. for the substance and is relative to water at 60°F.¹ The *scale of specific gravity* is scarcely ever used in practical work dealing with oils for lubrication or for fuels, and, instead, the *Baumé scale* is used. According to this scale, the specific gravity of water at 60°F. is called 10 degrees Baumé (10°Bé.) and liquids that are heavier than water have a gravity designation that is less than 10 degrees. In the oil industry, unfortunately, there are two Baumé gravity scales; one of these has been adopted by the U. S. Bureau of Standards and the other by the American Petroleum Institute. The latter is called also the Tagliabue Baumé scale. The U. S. Bureau of Standards Baumé scale is the one that is most generally used.

The relation between the Bureau of Standards Baumé scale and the ordinary specific-gravity scale for liquids lighter than water, which is the only case that applies to lubricating oils, is given by the following equation:

$$\text{Degrees Baumé (Bureau of Standards)} = \frac{140}{\text{specific gravity at } 60^{\circ}\text{F.}} - 130.$$

Similarly, the Petroleum Institute Baumé scale is given by a slightly different formula:

$$\text{Degrees Baumé (Petroleum Institute)} = \frac{141.5}{\text{specific gravity at } 60^{\circ}\text{F.}} - 131.5.$$

The table above corresponds to the values of the Bureau of Standards formula.

Specific gravity of a lubricating oil has no relation to its antifriction properties so that the terms "light," "medium," and "heavy" oil do

¹ Practically, of course, there is little difference between the density of water at 60°F. and at, say, 20°F. less or even 20°F. more.

not refer to specific gravities but to viscosity. These designations are misleading and go back to the time when it was thought that oils of high specific gravity had also high viscosity. In a general way, specific gravity of a lubricating oil indicates the type of crude oil from which the lubricant was made. For example, oils that have a density higher than 0.90 (Baumé less than 27 degrees), are likely to have been made from Texas crude oil, while those lubricating oils which have a density considerably less than 0.90 have probably been made from Pennsylvania crude oil.

The specific gravity of a lubricating oil has a useful commercial application for checking the grade of oil in a shipment. If an oil contract provides for the supplying of oil from a definite crude-oil base, and lubricating oil of a given density has been giving satisfaction, it is likely that all oil shipments from that source having the same density will be equally satisfactory.

If the temperature of the sample of an oil that is tested is not at 60°F., a correction is necessary. There are charts and tables that can be used for making this correction quite accurately.¹ For practical applications, it may be stated that to obtain standard conditions for lubricating oils, 0.05°Bé. should be added for each 1°F. that the temperature of the sample tested is above 60°F.; and similarly for kerosene and gasoline, 0.10°Bé. should be added for each 1°F. of temperature that the temperature of the sample is above 60°F.

Sometimes engineers will find it necessary to determine the specific gravity of liquids when suitable instruments are not available. Figure 300 shows a device which can be conveniently used in such cases. It consists of two U-tubes, connected together by rubber tubing as shown. Each U-tube is provided with the usual scale for observing the difference in level of the liquid in the tubes. One of these tubes is to be filled with **clean** distilled water (condensed steam) and the other with the liquid to be tested. When a slight pressure is produced in the tubes *A* and *B*, as, for example, by blowing with the mouth, the differences in the levels of the liquids in the two tubes are to be observed. The difference in level will be greatest, of course, in the tube having the lighter liquid. The ratio of the difference in level in the U-tube containing water to the difference in the level in the U-tube containing the liquid being tested is the specific gravity required. In the figure, if the U-tube *A* contains distilled water and *B* the liquid being tested, so that the water is

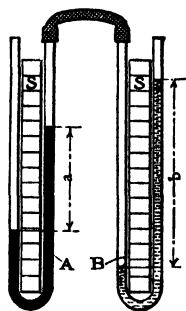


FIG. 300.—Simple device for determining specific gravity.

¹ DANFORTH, R. S., "Pressure Loss in Oil Pipe Lines," Kinney Manufacturing Co., Jamaica Plain, Boston, Mass., 1920.

displaced a inches and the other liquid b inches, then the specific gravity is $a \div b$. This sort of device can be made up anywhere where glass and rubber tubing are available.

Viscosity Test.—Every experienced user of oils knows how to determine roughly the relative fluidity, or what he calls the “body,” of an oil by rubbing it between his thumb and fingers. But such a test gives us no definite standard of comparison. In technical language this **relative fluidity** or “body” is called **relative viscosity**. It is generally accepted that the viscosity determination is a most important test of the lubricating qualities of an oil. In general, the higher the viscosity, the greater the lubricating value, but it should be observed, however, that care must be taken in interpreting the results of such tests because it does not follow, particularly for light, high-speed machinery, that a highly viscous lubricant like castor oil would lubricate a spindle for such machinery better than a “thin” sperm oil. In other words, as regards viscosity the lubricant must be selected with due consideration of the kind of work to be done.¹

A lubricant is applied to the bearings of machinery to keep the metallic surfaces moving over each other without coming into direct contact. Unless the surfaces are kept apart by some suitable fluid substance, the irregularities which exist on all surfaces, no matter how carefully they are made, will tend to interlock, and the friction of forcing or pushing them apart will generate heat. It is the function of a lubricant to flow between the closely fitting surfaces of a bearing and maintain always a thin fluid film to act as a cushion separating the solid particles of the bearing surfaces as well as to take up and carry away either by a direct flow or by vaporization a large part of the heat generated by friction.

Viscosimeters are instruments designed to determine the viscosity of oils. There is no generally accepted standard for such tests, as various types of instruments are used, and the results with different instruments will often vary considerably. Unless the names of the designer

¹ Further, in regard to the purely physical properties of oils, the viscosity determination is a test for the combined effects of both **cohesion** and **adhesion**. A good lubricant must have both of these properties to possess a satisfactory “body.” **Cohesion**, as it were, binds together the particles of oil. The more cohesion there is between the particles, the greater the pressure or force they will resist before separating. High cohesive value is therefore essential in lubricants intended for heavy machinery. **Adhesion** is the running mate of cohesion. It is the property of particles of one body adhering to particles of another body of different constitution. An oil with satisfactory adhesive qualities will stick or **adhere** closely to the surface of metallic bearings and thus assist the cohesive qualities by helping to resist the separation of the particles of the lubricant under pressure. A fluid having high cohesive properties and little adhesion would be a poor lubricant, as, for example, mercury; and conversely great adhesive qualities without those of cohesion, as in the case of water, alcohol, etc., are equally unsatisfactory. A good lubricant must have these two qualities jointly to make it viscous.

and maker of the instrument used and the amount and temperature of oil used are stated, results are almost meaningless.

It is generally considered necessary to make determinations of the viscosity of the so-called engine, dynamo, and machine oils at 70°F., which is considered the normal temperature in engine and dynamo rooms and in workshops, and again at about 120°, which is the temperature of a bearing that is running quite warm. Very often it will be observed that, of two samples of engine oil showing the same viscosity at 70°, one will test considerably higher at 120° than the other. Cylinder and gas-engine oils should be tested for viscosity at 100, 130, 160, and 210°F.; or if the cylinder oil is used in engines using superheated steam, or the gas-engine oil is for internal-combustion engines using extremely high compression, even higher limits of temperatures should be used in testing.¹ The best lubricant for a bearing under normal conditions may not do so well when heating begins. An oil which under ordinary conditions would be much too viscous for light, high-speed machinery, because it would be wasteful of power, is often most serviceable in a hot bearing when the engine oil ordinarily used vaporizes as soon as it enters and therefore serves no useful purpose. Heavy cylinder oil, however, since it vaporizes at a much higher temperature than the engine oil, will be heated only enough to make it thin and limpid without burning and will flow freely between the surfaces of the bearing, keeping the surfaces apart and at the same time conveying away much of the heat generated.

Types of Viscosimeters.—One of the simplest viscosimeters that is extensively used in America for determining the viscosity of oils is shown in Fig. 301. It is known as Scott's and consists of a metal cup with an orifice at the bottom, surrounded by an outer vessel also made of metal which for low-temperature determinations is filled with water. For tests at temperatures above the boiling point of water the vessel is to be filled with some kind of oil vaporizing at a temperature higher than that required for the tests to be made. The outer vessel is not circular but has a side extension shown on the right-hand side in the figure, under which a gas burner or an oil lamp can be placed for heating the liquid surrounding the inner cup containing the oil to be tested. By this means the oil in the cup can be heated uniformly to any desired temperature. The orifice is kept closed by means of a ball valve on a rod which extends up through the cover. The valve is lifted to allow the flow of oil through the orifice by pressing down with the finger on the end of the lever *A*. The handle *H* is provided for lifting out the oil cup so that it can be readily cleaned. A thermometer *T* registers the temperature of the oil. For draining the water or oil in the outer vessel

¹ When oils are to be tested at temperatures above about 200°F., usually the sample tested is heated by means of a bath of engine oil instead of water in the outer receptacle of a viscosimeter, particularly those of the "orifice" type where the direct application of a flame is impracticable.

a cock *C* is provided, and a glass cylindrical flask accurately graduated in cubic centimeters is supplied for measuring the discharge from the orifice.

To operate this apparatus, first fill the oil cup with about 200 cubic centimeters. In this apparatus it is necessary to start always with exactly the same amount of oil in the cup; otherwise there will be variable heads (pressures) on the orifice for different tests, introducing corresponding errors in the flow of the oil to be tested. Fill the outer vessel with water and heat the latter until the oil is at the required temperature for the test. Maintain this temperature constant for 2 or 3

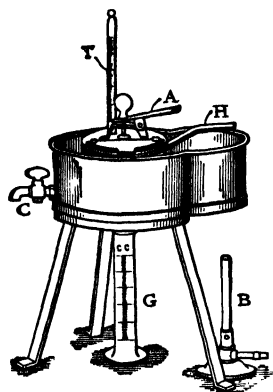


FIG. 301.—Simple type of viscosimeter.

minutes and then place a 50 cubic centimeter flask under the orifice and by removing the ball valve start the flow of oil observing the time as accurately as possible, preferably to the fraction of a second. When the level of the oil in the flask has reached the 50 cubic centimeter mark, close the ball valve and again observe the time. The number of seconds required for 50 cubic centimeters of a liquid to flow through the orifice is called the **time viscosity**. Usually instruments of this kind have the **water viscosity** marked on the name plate. This is the number of seconds required for 50 cubic centimeters of distilled water at 60°F. to be discharged from the orifice.

The water viscosity should be checked from time to time because there is sometimes a small accumulation of grease in the orifice which may remain unobserved and would reduce the actual or effective size of the orifice. Some basis for comparison of determinations of viscosity made by the various types of viscosimeters in which the discharge from an orifice is applied can be obtained by calculating what is called the **specific viscosity**. This is the time viscosity divided by the water viscosity. Thus if for a given apparatus the time viscosity of a liquid is 120 seconds and the water viscosity is 10, then the specific viscosity is 12.¹

Saybolt viscosimeter is in many respects a very satisfactory instrument for measuring "relative viscosity." Because of the ease by which it can be operated and the simplicity of its action, this instrument has been accepted by the U. S. Bureau of Standards, the American Society for Testing Materials, and the American Petroleum Institute as the standard

¹ Sometimes when oils of very different specific gravities are to be compared, a correction is applied to offset the difference in flow through the orifice, which causes the heavier oil to flow faster than a lighter one. Thus we obtain the **gravimetric viscosity** for comparison which is determined by dividing the specific viscosity by the specific gravity. For nearly all practical work the specific viscosity gives a sufficiently good basis for comparison of oils.

instrument for the measurement of the viscosity of oil. This instrument is shown in Fig. 302 and consists in its essential parts of a cup which has a small orifice at the bottom for the discharge of the oil, which is put in at the top of the cup and filled to a standard level.¹

The Saybolt universal viscosimeter is made entirely of metal. An oil tube *O* is provided at the top of the instrument with an overflow cup *C*, and at the bottom with a small outlet orifice through which the oil to be tested flows into a flask *F*, the capacity of which, filled to the mark on its neck, is 60 cubic centimeters. The lower end of the outlet tube containing the orifice is enclosed by a larger tube in which a cork *K* is to be inserted that forms a closed air chamber, the cork preventing the flow of oil through the orifice until the cork is removed and the test is started. The oil tube *O* is set in a jacket which may be heated by steam or by water that is heated by a gas flame or by an immersed electric heater. The steam or hot-water pipe *W* used for heating the baths may also be used to carry cold water through the pipe to reduce the temperature of the bath, if it is too high for the purpose of the test that is to be made.

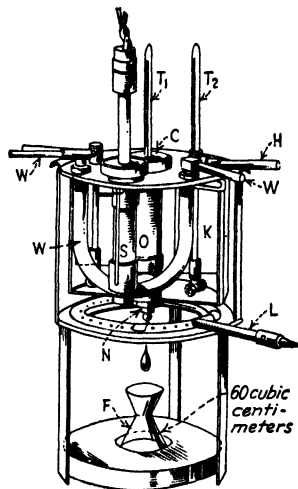


FIG. 302.—Saybolt universal viscosimeter.

Saybolt Furol Viscosimeter.—When the viscosity of oily, viscous petroleum products like fuel oil or road oil is to be determined, the Saybolt standard viscosimeter is not suitable and the so-called Saybolt Furol viscosimeter is used instead, the Furol type having an orifice that delivers oil volume at ten times the rate of the standard type. When the Furol instrument is used, viscosity results are usually stated as "Saybolt Furol" at 70 or 100°F.

Redwood's and Engler's viscosimeters are practically the same as Scott's (Fig. 301). Redwood's is largely used in England and Engler's is officially adopted by the German government. None of the types described have orifices of exactly the same size.

The viscosity scales of all industrial types of viscosimeters using the efflux (out-flow) principle have arbitrary values. A chart for

¹ The only source available for driving the oil through the orifice in the Saybolt instrument is that of gravity and the force holding the oil back in the cup is its viscosity or what may be called its internal resistance. The time required for the standard quantity of oil to flow through the orifice will depend on both the viscosity and the density of the oil. The Saybolt instrument measures a unit comparable to the absolute viscosity divided by the density, which is called the **kinematic viscosity**. There are a number of other viscosimeters applying the same principle of the orifice or the short tube, and in all such instruments the time of flow must be corrected to take into account the density of the oil.

making conversions from the readings of one type of viscosimeter to another type which has been prepared and published by the Society of Automotive Engineers¹ is given in Fig. 303. It will be noted that the

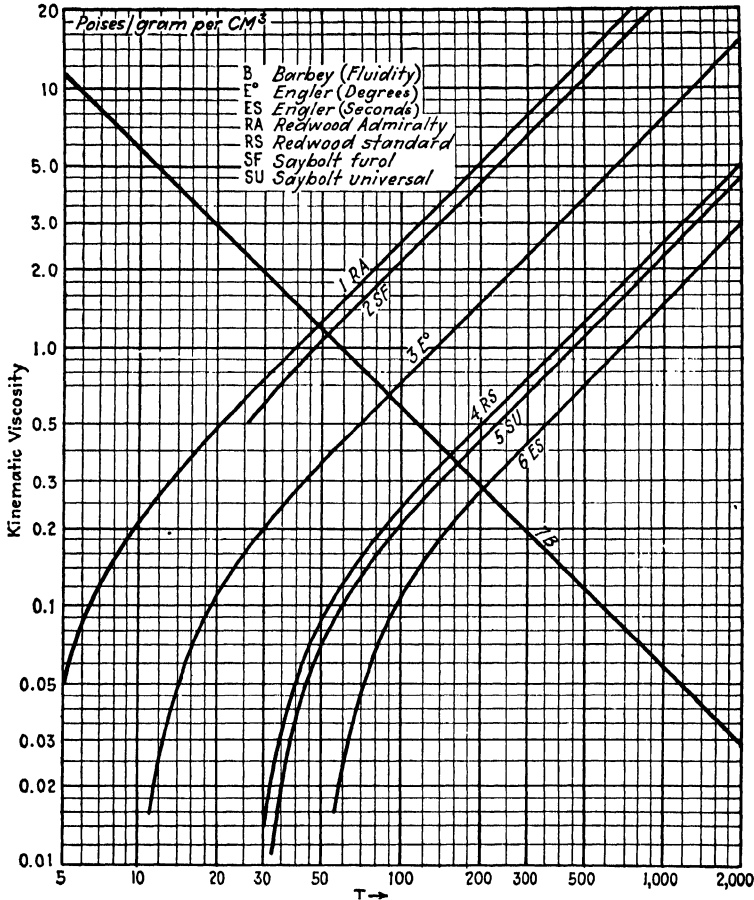


FIG. 303.—Conversion chart for comparison of various viscosimeters.
 For curves 1, 2, 4, 5, and 6, T = time in seconds.
 For curve 3, $T = 10 \times$ Engler degrees (6 Engler degrees = time in seconds).
 For curve 7, time = Barbey fluidity (rate of flow, cu. cm. per hour).

curves in this figure have been plotted on *logarithmic paper* (page 503). When such data are plotted in this way, nearly straight lines are obtained, except for very thin oils having Saybolt standard viscosity of less than 140. This fact can be used for checking the results of viscosimeter tests. All the curves in the chart have a drooping tendency at the left-hand side of the curve sheet.

¹ *Jour. S. A. E.*, Vol. 10, p. 31, 1922.

According to standard Saybolt viscosimeter tests, types of lubricating oils have been classified as follows:

Light body.....	95 to 115 seconds at 210°F.
Medium body.....	116 to 135 seconds at 210°F.
Heavy body.....	136 to 170 seconds at 210°F.

In the efflux type of viscosimeter, application is made of the fact that, when a liquid flows through an orifice, its rate of flow increases as the internal friction of the liquid decreases. In the application of this principle, therefore, it is possible to compare lubricating oils for the purpose of determining the variation in the same oil for different temperatures.

Device for Absolute Viscosity.—Frictional resistance and bearing temperature are in proportion to the absolute viscosities, so that all readings or observations for many of these instruments must be reduced to absolute units before they can be useful for comparison.

One of the methods used in measuring absolute viscosity is based on the time required for a definite quantity of oil to flow through a capillary tube. Very viscous oils will flow through such a tube more slowly than thin oils, owing to the greater resistance in their passage through the tube. Figure 304 shows the design of such a viscosimeter.

The method of making the measurement is as follows: The clean and dry instrument is filled by pressure in the right limb from *H* to *A* with the oil to be measured, the surplus oil overflowing into the trap at *A*. The oil is introduced by means of a pipette. The left limb is connected to a tank filled with air under pressure, which can be measured on a water manometer. Then when the right limb is connected with the air, the time of falling of the meniscus from the aperture at *B* to the one at *D* is taken as the time of flow.

If the time of flow *t*, the pressure on the oil in the left limb *p*, the density of the oil *d*, and the two constants of the instrument *c* and *c'* are known, it is possible to calculate the **absolute viscosity** for the temperature of observation from the formula

$$\text{Absolute viscosity} = cpt - \frac{c'd}{t}.$$

Saybolt and Absolute Viscosity Compared.—The number of seconds required for 60 cubic centimeters of a sample of oil to flow through the orifice of a Saybolt viscosimeter is called the Saybolt viscosity of the oil. Thus, when it is stated that a sample of oil has a Saybolt viscosity of 220, it takes 220 seconds for the standard quantity of oil at 68°F. to flow through the orifice in the instrument. A long time is required for the

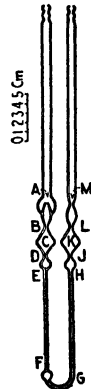


FIG. 304.—
Device for absolute viscosity determination.

standard quantity of a heavy-bodied or viscous oil to flow through the orifice of the instrument, and a sample of oil of this kind will have a high reading as expressed in Saybolt viscosity. The relation between Saybolt viscosity and absolute viscosity is shown in a chart which was prepared by one of the leading oil companies.¹ It will be noticed that for Saybolt readings of about 200 seconds, the absolute viscosity is about one-fifth of the Saybolt viscosity. For example, a Saybolt viscosity of 220 seconds is equivalent to about 44 in units of absolute viscosity.

When calculations for the *comparison of frictional resistance in bearings* are made, for example, the Saybolt readings should always be converted to the equivalent absolute viscosity. The conversion may be made by the use of the following formula as determined by the U. S. Bureau of Standards.²

$$\text{Absolute viscosity units} = 0.00220t - \frac{1.80}{t}.$$

Since the readings of a Saybolt instrument are affected by the density of the oil, it is obvious that, if two kinds of oil are being tested which have different specific gravities but have the same Saybolt viscosity in seconds, their absolute viscosities will still be different. For example, if one kind of oil has a specific gravity test of 30°Bé. and another oil has a specific gravity of 25°Bé., the oil having the lower value in terms of Baumé gravity will actually have the greater absolute viscosity. In this case the difference would amount to about 7 per cent.

Attempts have been made to devise relatively simple instruments for determining the viscosity of oil in power plants, but without very much success, as it is almost impossible to maintain a constant temperature of the oil while the test is being made. This difficulty applies to such instruments of this comparatively simple type as those having, for example, a steel ball dropping through a tube filled with oil, a bubble of air rising through a tube filled with oil, or in which the drops from the end of a pencil or from the orifice of a small engine lubricator are counted.

In making tests for viscosity it is difficult for some to realize the importance of constant and practically the same temperature when both the sample of oil and the water are passed through the instrument. Few people realize that water is one-fourth as viscous at 190°F. as at 50°F. Oils are, of course, much more noticeably viscous at low temperatures than when heated, as they are relatively so much more viscous and thicker than water at ordinary room temperatures, and the rate of change of viscosity with increase in temperature is much more rapid with oil than with water. Because the viscosities of oil and of water change so much with the temperature, it is always necessary in stating determination

¹ This chart as published in *Lubrication*, Vol. 7, No. 5, may be obtained by application to the Texas Company, 17 Battery Place, New York City.

² See *Tech. Paper* 112, U. S. Bur. Standards.

of viscosity to specify the temperature at which the viscosity tests were made and the instrument used, so that accurate comparison may be made. The standard temperatures for viscosity measurements are as follows:

Turbine engine and machine oil.	100°F.
Steam cylinder and gear oils.	210°F.
Internal-combustion cylinder oils (automobile and Diesel engines) . . .	100 and 210°F.

Cloud Test of Lubricating Oil.—Lubricants are frequently tested to determine the temperature at which paraffin wax and similar sub-

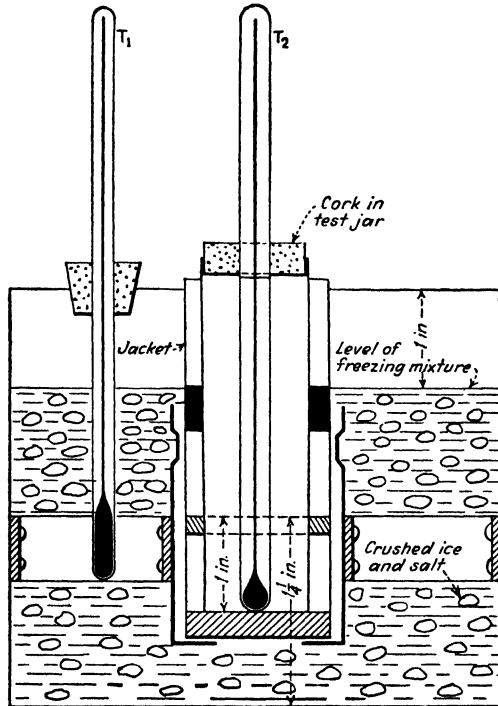


FIG. 305.—Device for cloud test and pour or cold test.

stances found in oil crystallize and separate from the oil. A test of this kind is called the *cloud test*. A simple apparatus for making this test of oil is shown in Fig. 305. It consists of a glass jar preferably a little more than 1 inch in inside diameter and about 5 inches high. In making the cloud test, the oil is poured into the glass jar until the level is about $1\frac{1}{4}$ inches above the bottom, the depth of oil being at least sufficient to permit reading the scale of the thermometer at the level of the oil. For this test a special thermometer is needed which has a bulb that is only about $\frac{3}{8}$ inch long. This thermometer is inserted into the jar through a tight-fitting cork which will hold it in a central position with the lower end of the bulb $\frac{1}{2}$ inch from the bottom of the jar. The jar with its

thermometer properly in place is then placed in a metal or glass jacket container 5 or 6 inches high having an inside diameter at least 1 inch larger than the outside diameter of the test jar. A disk of felt or cork about $\frac{1}{2}$ inch thick is placed in the bottom of the jacket container and then the test jar containing the oil is placed centrally on the felt or cork disk. The jacket between the jar and the container is then filled with a refrigerating or freezing mixture which may consist of ice mixed with either calcium chloride or sodium chloride. For oils congealing or solidifying from 15 to 35°F., a satisfactory freezing mixture is made of cracked ice and sodium chloride in the portions of 1 part of salt to 20 parts by volume of cracked ice. For oils congealing at temperatures from -5 to 15°F., a larger percentage of salt is required than for the higher temperature. In this latter case, there should be about 1 part of salt to from 3 to 5 parts of cracked ice. For still lower temperatures, as, for example, those as low as -20 to -25°F., a mixture of 1 part of *calcium* chloride to 3 parts of cracked ice may be used.

The cloud test is usually made by putting into the test jar oil that has been heated in another vessel to a little more than 150°F. and has been cooled in the air to the "room" temperature. The cooling is continued in the refrigerating bath, the jar being taken out at every reduction of 2°F. in its temperature below the point where congealing of the oil may be expected to begin. Care must be taken when removing the test jar not to move the thermometer, as its movement would hinder the congealing of the oil. When the lower half of the sample becomes opaque through chilling, the temperature of the thermometer in the oil should be taken and recorded as this temperature is the cloud test temperature of the oil.

Pour or Cold Test for Congealing of Oils.—There is often a great difference in the temperature at which lubricating oils which have much the same characteristics at room temperatures become viscous when they get very cold. In every sample of oil there are a number of different hydrocarbons and each of these hydrocarbons has its own particular freezing point; and when a number of the hydrocarbons have freezing points at higher temperatures than most of those in the oil, they will separate out and float on the surface of the oil. If the hydrocarbons which freeze at relatively high temperatures are present in sufficient quantities in the oil, it will cause the entire mass of oil to congeal. Oils which have a paraffin base or, in other words, contain a considerable amount of paraffin wax are likely to congeal at a much higher temperature than oils which have an asphalt base and contain no paraffin wax.

This is an especially important test for oils used in automobiles in winter service. The same apparatus that is used for the cloud test of oil may be used to determine the temperature at which a sample of oil will just flow, this temperature being the *pour test* temperature of the oil. In many cases the pour test of an oil may be taken immediately after the

cloud-test temperature has been determined, as in most cases the cloud test has a higher temperature than the pour test. In any case, in making the pour-test determination after the sample of oil has been cooled to about the expected pour-test temperature, for every reduction of 5°F. in temperature, the jar containing the sample of oil is taken from the jacket containing the freezing mixture and is held in an inclined position not more than 10 seconds. The cooling is continued until the bottle can be held at an angle of 45 degrees without causing the oil in the jar to flow. When this point is reached, the previous test point (5°F. less than the point of solidification of the oil) is taken as the "pour-test" temperature of the oil. As a rule, the pour test should be completed in about $\frac{1}{2}$ hour.

The simple method of making this test in a power plant is to fill a 4-ounce bottle with a sample of oil and place it where it will be exposed to low temperature, for example, in a brine tank of a refrigerating system. If the sample is frozen solid after an exposure for 24 hours, the thermometer should be put into the neck of the bottle so that the temperature may be observed as the bottle is heated. The pour test shows the temperature at which the oil in the bottle begins to flow. This method of testing requires that the oil shall be frozen without stirring or other disturbances. This test is, therefore, not exactly useful for testing the suitability of the oil for use, for example, in a ring-oil bearing or in the clearance space between a shaft and its bearings which are being agitated whenever the shaft moves and the tendency is to prevent oil from congealing at the pour test. This temperature may be as much as 15 to 25°F. lower than the actual congealing point of the pour test. On the other hand, the oil in ring-oil bearings which have been warmed up by friction and by the heat conducted from the engine, if cooled down to any great extent after the engine is shut down, may congeal at a higher temperature than the pour test. The pour test made in a 4-ounce bottle as described above, or in a test tube as prescribed by the A.S.T.M., does not always indicate the temperature at which the oil will flow out of a barrel or a wide-mouthed can. As a rule, samples of oil can actually be poured at temperatures below the temperature of the pour test.

Carbon-residue Test of Lubricating Oil—Conradson's Test.—If a lubricating oil is heated to a high temperature in an enclosure where there is a limited supply of air, the greater part of the oil will be distilled and there will remain a residue of carbon. The amount of this carbon residue in a lubricating oil is an indication of the extent of its decomposition when used for high-temperature lubrication. Carbon-residue determinations are generally made by the method introduced by Conradson, and the apparatus shown in Figs. 306 and 307 is used. This apparatus consists of the following equipment:

1. Porcelain crucible *P* (wide form) glazed throughout, 25 to 26 cubic centimeter capacity, 46 millimeters in diameter.

2. Skidmore iron crucible *S*, 45 cubic centimeters ($1\frac{1}{2}$ -ounces) capacity, 65 millimeters in diameter, 37 to 39 millimeters high with cover, without delivery tubes and one opening closed.

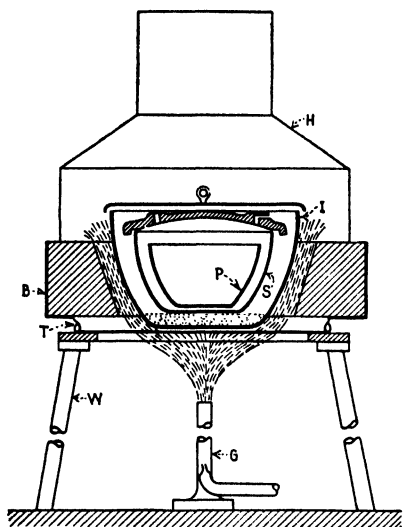


FIG. 306.—Conradson carbon residue apparatus.

3. Wrought-iron crucible *I* with cover, about 180 cubic centimeters capacity, 80 millimeters in diameter, 58 to 60 millimeters high. At the bottom of this crucible a layer of sand is placed about 10 millimeters deep or sufficient to bring the Skidmore crucible with cover on, nearly to the top of the wrought-iron crucible *I*.

4. Triangle *T*, pipe stem covered, with a projection so as to allow the flame to reach the crucible on all sides.

5. Sheet-iron or asbestos hood *H* provided with a chimney about 2 to $2\frac{1}{2}$ inches high, $2\frac{1}{8}$ to $2\frac{1}{4}$ inches in diameter, to distribute the heat uniformly during the process.

6. Asbestos or hollow sheet-iron block *B*, 6 to 7 inches square, $1\frac{1}{4}$ to $1\frac{1}{2}$ inches high, provided with opening in center $3\frac{1}{4}$ inches in diameter at the bottom, and $3\frac{1}{2}$ inches in diameter at the top.

Method of Testing for Carbon Residue.—In testing lubricating oil in Conradson's apparatus (Fig. 306), approximately 10 grams of the oil

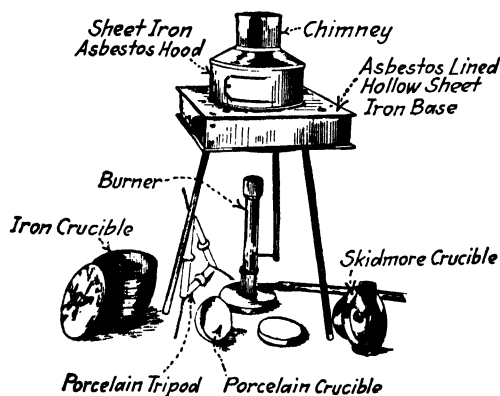


FIG. 307.—Apparatus for determining carbon residue.

are weighed in the porcelain crucible *P*, which is then placed within the Skidmore crucible *S*, and the two crucibles (one inside the other) are put into the larger iron crucible *I*, care being taken that the Skidmore crucible

is set in the center of the iron crucible. As shown in the figure, suitable covers are required over the Skidmore and the iron crucibles. The set of crucibles thus concentrically arranged are then placed upon a triangle *T* (pipe stem covered), which rests on a suitable iron chemist's stand *W*, the crucibles being surrounded by the asbestos or hollow sheet iron block *B*, which also rests on the triangle *T*. The stand and crucibles are then covered with the sheet-iron or asbestos hood *H*, which serves to distribute evenly the heat that is to be supplied during the test. The asbestos or sheet-iron block *B* serves as a shield for the flame from the gas burner *G* and guides the flame around the iron crucible *I*.

Heat is applied to the apparatus with a Bunsen or similar gas burner with a hot high flame which surrounds the iron crucible *I*, as shown in the figure, until vapors from the oil start to ignite over the crucible (usually from 4 to 7 minutes). When the vapors from the oil begin to ignite, the gas flame is lowered (to a height of 2 or 3 inches above the burner), and the first appearance of vapors must be carefully watched. After this period of "strong" heating, in order that the oil *vapor* flame may not get too high, and to prevent the boiling of the oil in the crucible, this low height of the gas flame is required. The vapor flame should never exceed a height of 3 inches above the chimney *H* and should be kept at an average of about 2 inches above the chimney during the period in which vapor escapes from the surface of the oil. After the vapor ceases to come off (as shown by inability to ignite it), the heat from the gas burner is increased to its maximum amount (about as at the start of the test) and kept at this high value for 5 minutes. The bottom of the iron crucible *I* during this period of maximum heating should be red hot. At the end of this 5-minute period of high heat, the gas flame is put out and the crucibles are allowed to cool (with the chimney removed) for 5 minutes, after which the covers of the crucibles *S* and *I* are removed, and the porcelain crucible *P* is taken out and put into a desiccator where cooling is continued. When cooled to room temperature, the crucible with its residue is accurately weighed. The entire test should be finished in about 30 minutes if the heat is properly regulated. The time, however, will depend somewhat on the kind of oil, as ordinarily a rather thin, low flash-point oil will require less time for this test than a heavy, thick, high flash-point oil.

In order to get accurate results, the dimensions of the apparatus used must be as nearly as possible the same as those specified in this description. Special precautions must be taken to observe the first appearance of vapors. In order to make this observation as accurate as possible, the gas burner may occasionally be momentarily removed to facilitate this observation. If, at any time during the test the vapor from the oil exceeds the 3-inch specification above the chimney of the hood, the gas burner may be removed for a short interval until the size of the gas flame

can be reduced sufficiently to keep the vapor flame about 2 inches above the top of the chimney.

Many specialists in lubrication question the correctness of results from this test, for the reason that it does not reproduce the conditions in practice that cause the deposit of a carbon residue. In the actual use of oil for lubrication in an engine cylinder, the oil is spread out in thin films on hot metal and is exposed to highly heated, burning gases which, in most cases, will have an oxidizing effect. The amount of oil that gets into the combustion space of an engine cylinder depends on the mechanical fit of the piston and the piston rings, as well as on the viscosity of the oil. A "lean" mixture of oil fuel and air in an engine cylinder will have for combustion an excess of oxygen to burn up the oil, while, on the other hand, a "rich" mixture of oil fuel and air is less likely to have an excess of oxygen; and in the absence of the oxidizing atmosphere, the oil that passes into the combustion space by leaking between the piston rings and the cylinder wall cannot burn completely and will leave a carbon deposit in the engine cylinder. These facts are stated for the purpose of making it clear that the carbon-residue test may give actually very little information in regard to the performance of an oil in an engine cylinder.

Water and Sediment. Lubricating Oils. Centrifuge Tests.¹—Lubricating oils made by reputable refiners are free from water and sediment before they are used in machinery, so that tests for water and sediment in oil are intended mainly for use in connection with *oils that have been used*. These tests are especially useful in the case of used oils from circulating systems of steam turbines and internal-combustion engines in large plants where it is profitable to recondition oil so that it may be used again and again for lubricating purposes.

Oils used for the lubrication of steam turbines are, as a rule, contaminated very quickly by dilution with water which enters the bearings of a turbine as steam leakage. Lubricating oil used for this service can, therefore, be made available to be used again, if water and sludge are removed. Reconditioned oil should, however, be tested before being used again by some approved method like the one given here, in order to determine if practically all of the water and sediment have been removed.

Lubricating oils used in automobile engines are also deteriorated by dilution, but by the presence of gasoline instead of water. Oils that are diluted with gasoline are not usually reconditioned. It is sometimes necessary, however, to make tests of the lubricating oil removed from automobile-engine crank cases for a sediment determination, the sediment

¹ For more details of testing procedure, see Shoop and Tuve, "Mechanical Engineering Laboratory Practice," p. 154. Centrifuges and centrifuge tubes may be obtained from E. F. Mahady Company, Boston, Mass. Any convenient frame with whirling attachments of the required diameter that is suitable for the attachment of standard centrifuges may be used for this test.

being due mostly to sand and dirt that get into the engine through the breather pipe and the carburetor and to bits of metal that have been worn from bearing surfaces and are corroded from those parts where water is present. It is often stated that the amount of sediment found in samples of used automobile engine oils is likely to be an indication of the carbon-residue properties of the oil. In this connection, however, it should be kept in mind that carbon-residue determinations will depend on the "richness" of the fuel mixture in the same way as explained when describing the method of obtaining the carbon residue in lubricating oils by direct heating (page 536). For automobile engines used largely on roads built of soft materials, especially gravel, the amount of sediment determined by this test will be very much greater for the same engine conditions than for automobiles that are used almost entirely on hard roads.

Centrifugal Method of Testing Lubricating Oils.—The method to be described of testing lubricating oils by the use of a centrifuge, for the purpose of obtaining the amount of contained water and sediment, has had quite general commercial acceptance, although admittedly it is not very accurate, especially as to the amount of water, the results being in

nearly all cases less than the actual water content. Nevertheless, as a general rule, it is needless to expect great accuracy from this method of testing, for the reason that there is always uncertainty as to whether the sample obtained for testing is really representative. Because of the difficulty in obtaining a good sample, it should be clear that the sampling should be done as carefully as possible. The centrifuge used for such testing is relatively simple in its parts and need not be designed with as great care as a similar apparatus intended for much higher speeds. The centrifuge for testing the water and sediment in lubricating oils is usually made between 15 and 17 inches in diameter (tip to tip) of the centrifuge tips and revolves at a speed of about 1,500 revolutions per minute. The rotating member of the centrifuge may, however, have any other diameter which will give equal tip speed. Typical centrifuge tubes are shown in Fig. 308.

Centrifuge tests of lubricating oils are made by filling each of the centrifuges with a mixture of benzol (90 per cent pure) and of the oil to be

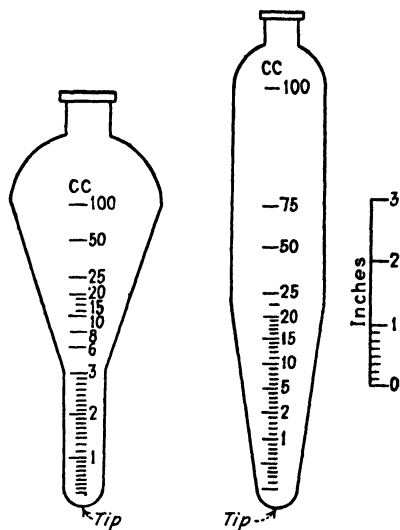


FIG. 308.—Typical centrifuges.

tested, with 50 cubic centimeters of the benzol and 50 cubic centimeters of the oil. After the mixture of benzol and oil has been poured into each of the centrifuges, they are to be tightly corked or stoppered and then shaken vigorously until the contents are thoroughly mixed. The filled centrifuges are then to be placed in a water bath at 100°F. and are to be immersed to the 100 cubic centimeter mark for 10 minutes. After this immersion, the centrifuge tubes (usually two) are placed in their proper positions on diametrically opposite sides of the centrifugal frame to be whirled at the required speed for 10 minutes. At the end of this period, the centrifuge frame is to be stopped, the centrifuge tubes removed, and a reading taken of the volume of water and of sediment at the bottom of each tube. After these readings have been taken and recorded, the centrifuge tubes are again placed in the centrifuge frame and whirled for an additional 10 minutes, and, at the end of this time, when the centrifuge frame is stopped, readings of water and of sediment are again taken and recorded. This operation is repeated until the combined volume of water and sediment in each tube remains constant for three successive readings. Usually, only four sets of observations are needed.

Emulsification Test of Lubricating Oils.—In some kinds of service, emulsification of oil is a desirable quality, while in other kinds of service it is undesirable.

For example, lubricating oils intended for circulation in steam-turbine bearings should not emulsify. The so-called mineral lubricating oils are more resistant to emulsification than "compounded" petroleum oils that are mixed with animal and vegetable oils, such as wool, grease, tallow, or lard oil. Emulsification tests may be made with the object of determining: (1) the degree of emulsification of oil; or (2) the rapidity of separation when there is little or no emulsification. There is, however, no standard emulsification test. Herschel, a specialist in the U. S. Bureau of Standards, has proposed the following method, which has received some acceptance. Into a glass container of about 100 cubic centimeter capacity, 20 cubic centimeters of the oil to be tested and 40 cubic centimeters of water are poured. Insert into this glass container a paddle about 90 millimeters long, 20 millimeters wide, and 1.5 millimeters thick. The glass container is then placed in a water bath where the temperature is maintained at approximately 130°F. This paddle is to be set up vertically in the glass container and with a suitable turning device is rotated at 1,500 revolutions per minute for 5 minutes. Record is made of the time required for separation of the water from the oil by taking readings at intervals of about 1 minute. The demulsibility figure D in the following formula is calculated from the rate at which the water separates from the oil per hour and is expressed by $D = 60v \div t$, in which v is the volume of oil separated out in cubic centimeters, and t is the corresponding time in minutes. Thus, if all the oil in the sample (20 cubic centimeters) sepa-

rates from the water in 60 seconds, then the demulsibility figure equals $60 \times 20 \div 1$ or 1,200. On the other hand, if the sample of oil emulsifies so badly that only 15 cubic centimeters of it separate from the water in 15 minutes, the demulsibility figure $D = 60 \times 15 \div 15$ or 60.

Compounded or "loaded" lubricating oils made by the addition of wool, grease, tallow, or lard oils to rather heavy types of petroleum oils are suitable for some kinds of steam-engine service, especially for engine-cylinder lubrication. When compounded oils are subjected to high temperature and moisture as in the cylinder of a steam engine that does not use highly superheated steam, the oil will readily emulsify and adhere to the wet cylinder walls, thus improving the effectiveness of the petroleum lubricant for engine operation. On the other hand, a pure mineral oil would not adhere in the same way and would freely remove itself from the walls of an engine cylinder. Compounded steam cylinder oils contain usually from 5 to 10 per cent by weight of animal or vegetable oils, or both. It should be noted, also, in connection with compounding of engine oils, that a vegetable oil (castor oil) is frequently used for mixing with mineral oil for airplane service.

A diluted lubricating oil is sometimes most advantageously used, for example, in some automobile engines (gasoline dilution), where splash lubrication in large amounts is necessary for cooling the engine parts which would not be reached by the oil distribution that would be possible without dilution. This is also true of some small vertical steam engines that are lubricated almost entirely by crank-case "splash." Some types of steam engines that are lubricated largely from the crank case will actually operate more successfully with a mixture of oil and water for lubrication than with oil alone. For such mixing with water, however, the oil must be practically free from any emulsifying action.

Flash-point Test and Fire (Burning-point) Test of Oil.—At one time very important parts of the specifications for oil were the flash point and the fire test. The flash point indicates the temperature at which an oil gives off vapors in such amount that, in combination with air, they form an inflammable mixture. Formerly the flash point was considered especially important because of its relation to the fire hazard in the storage and shipping of oils. When there was relatively little gasoline stored and shipped, the flash-point test of an oil was probably much more important than it is now, when gasoline is being shipped and stored in so much larger amounts than any other petroleum product.

The burning point of an oil is the temperature at which the oil ignites and continues to burn. The burning point is usually from 5 to 20°F. higher than the flash point.¹ The value of an oil as a lubricant has no relation to its flash or its fire-test or burning point.

¹ Lubricating oil will not take fire and burn by heat application alone, until it is heated to a temperature that is considerably higher than this fire test.

A lubricating oil will not ignite itself merely because its temperature is slightly above the flash or burning point. Until a considerably higher temperature is reached, external ignition is necessary. Up to the end

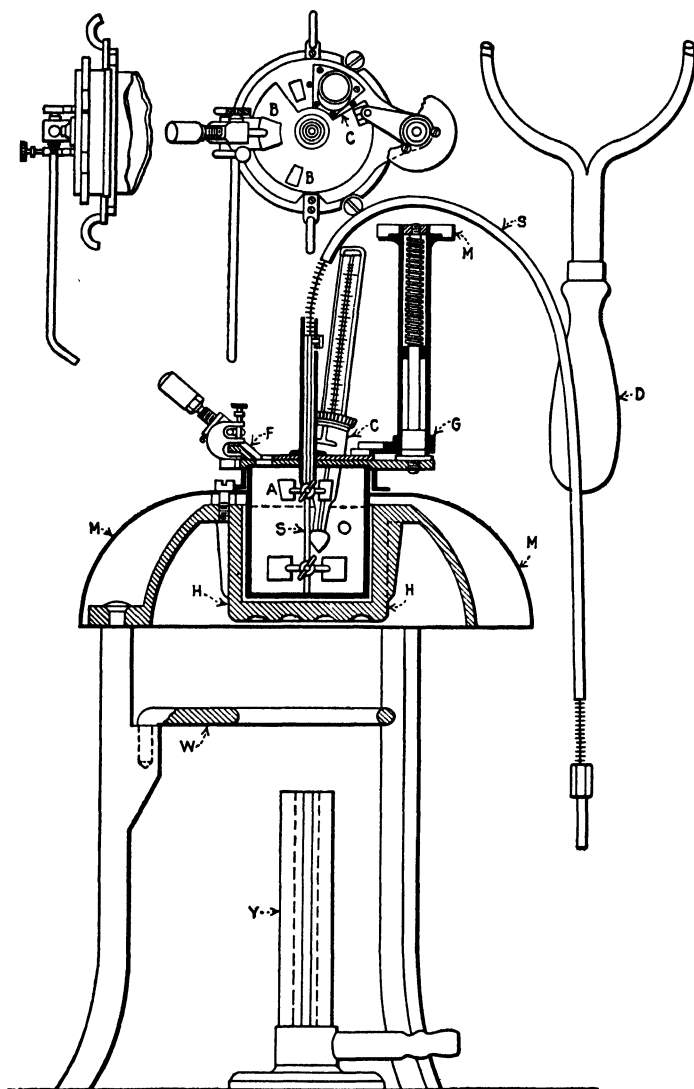


FIG. 309.—Pensky-Martens closed oil tester

of the last century, many lubricating oils had flash or burning points under 300°F., so that a hot bearing on an engine or machine was a potentially dangerous fire hazard. In recent years, however, lubricating oils have been so much improved in the refining processes that the flash and

burning points of good quality oil have been raised, so that they are now nearer 400°F. than 300°F.

Apparatus for Determining Flash Point and Fire Test of Oil.—There are two kinds of apparatus, that is, (1) open type and (2) closed type, that are used for flash-point and fire-test determinations of oil. Among the closed type of apparatus for these determinations, the *Pensky-Martens instrument is now generally accepted as the standard*. This apparatus is shown in considerable detail in Fig. 309. The Cleveland Tester (Fig. 310) is the type of open tester that is most used. The Pensky-Martens tester is the more accurate of the two, but the Cleveland open tester gives sufficiently accurate results for most practical work.

The Pensky-Martens Closed Tester.—The oil tester of the closed type shown in Fig. 309, called Pensky-Martens, consists of an oil container *O*, which is placed in a metal heating vessel *H*, provided with a mantle or covering *M* to protect the heating vessel *H* from heat loss caused by radiation. The oil cup *O* is closed by a closely fitting brass lid. Through the center of the lid passes the stirring mechanism, consisting of a slender shaft *S* carrying two propellers used for stirring, one of them rotating in oil, and the other in the vapor space just under the lid. The stirring mechanism is operated by the handle *D*. The lid has four other openings, the position and size of which are accurately made (see top view in figure). One of these openings is circular and the thermometer collar *C* is concentric with it, the thermometer being inclined at a convenient angle. The other three openings are opened and closed by a slide valve *B*, which is controlled by a spring and lever device *G*, which is rotated by turning the milled head *M*. One of these openings (middle) is used for inserting the testing flame, and the others for admitting air required for combustion. The test flame is controlled mechanically by the operation of the slide valve *B* which inserts the test flame into the space above the oil. With every turn of the milled nut *M*, the spring and lever device *G* is moved, so that the middle opening in the slide valve *B* comes opposite the orifice in the lid; and, at the same time a very small flame burning at the movable jet *F* is moved so that it comes very close to the surface of the oil that is being tested. This test-flame burner should have a tip opening of $\frac{1}{32}$ inch in diameter. A pilot-light burner is maintained near the test-flame burner, so that every time the slide valve *B* is open the test flame is automatically lighted.

Test Procedure.—In order to start a test with the Pensky-Martens tester, the cup *H* is filled to a definitely standardized marked level with

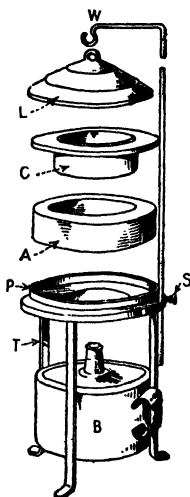


Fig. 310.—Cleveland oil tester.

the oil to be tested. The lid is then put in place, and the oil is heated somewhat rapidly at first, until its temperature is about 50°F. below the expected flash point. The temperature of the oil is then increased slowly, about 10°F. per minute, the temperature rise being conveniently controlled by the use of the wire gage *W*, as shown in the figure, just above the Bunsen gas burner *Y*. During this latter heating, the milled head *M* is turned somewhat slowly, so that the test flame is tilted about once a minute into the oil cup *O*. The temperature at which the first slight explosion due to the vapor ignition is observed is to be recorded as the flash point of the oil. This instrument is only a flash-point tester.

Cleveland Open Tester.—An open type of oil tester which is used a great deal in the commercial testing of lubricating oils is shown in Fig. 310. It is a simple device consisting of only six essential parts, which are (1) a brass cup *C*, $2\frac{1}{2}$ inches inside diameter, and $1\frac{1}{8}$ inches deep. The side wall of the cup is $\frac{3}{16}$ inch thick, while the bottom is $\frac{1}{16}$ inch thinner;¹ (2) a brass plate *P* with a circular depression to fit the rim of the cup; (3) a hard asbestos plate $\frac{1}{4}$ inch thick, with a hole to fit the cup; (4) a brass lid *L* which fits the cup; (5) a tripod *T* which supports the brass plate; (6) a nitrogen-filled etched-stem glass thermometer having a bulb approximately $\frac{1}{2}$ inch long, which is not larger in diameter than the stem. Heat is applied to the oil cup by the use of a Bunsen gas burner which should be protected from air drafts by a suitable shield which may extend up to the level of the asbestos plate *A*.

Test Procedure.—In order to make a test of oil with the Cleveland open tester, the cup *C* is filled up to a standardized mark on the inside of the cup, which is about $\frac{3}{8}$ inch below its upper edge. The asbestos ring *A* is then placed around the oil cup *C* and the bottom of the cup is inserted into the depression in the plate *P*. The cup, the plate *A*, and the plate *P* are then placed on the tripod *T*. In this position, the cup is heated by the flame from a suitable burner which is adjusted to heat the oil at a uniform rate of 10°F. per minute. The thermometer for observing the temperature of the oil may be conveniently suspended from a wire support *W* adjusted in height by means of a thumb screw *S* fastened to the side of the tripod *T*. The thermometer must be so located that its bulb is halfway between the center and the side of the cup and is completely immersed in the oil, but it is lower and must be as nearly as possible about $\frac{1}{2}$ inch from the inside bottom of the cup. If the special flash-test thermometer provided by the manufacturers of this equipment is not used, correction for emergence of the stem (page 45) of the thermometer must be made. It is necessary to be careful in making this test to be sure that the temperature of the oil is continually rising, as any recession at about the flash point or the burning point would

¹ Exact dimensions of this equipment are given in Diedrichs and Andre, "Experimental Mechanical Engineering," Vol. I, p. 919, 1930.

invalidate the results. In other words, the temperature of the oil must be kept steadily increasing at the specified rate throughout the test, except that in the early part a higher rate of heating (up to above 50°F. below the expected flash point) is allowable. The test flame should preferably be from a very small gas burner with a tip orifice not much larger than the one in the Pensky-Martens flash tester; but instead of the gas flame a long splint of wood or a wax taper may be satisfactorily used. The flame of whatever kind should not be more than about $\frac{1}{4}$ inch in length and should be drawn across the cup in the horizontal plane of its top edge. The test flame should be applied in this way at 5°F. intervals throughout the test. The temperature at which the first flash of vapor combustion takes place near the surface of the oil is to be noted and recorded as the flash temperature, which means that this is the point at which the oil liberates its vapor at a sufficient rate so that it may be ignited. The flash point must not, however, be confused with a bluish halo that sometimes surrounds the test flame at a temperature much below the flash point. Obviously, air drafts must be carefully avoided in a testing apparatus of this kind, and it is a good precaution to partially darken the room where a test is being made, so that the first flash may be observed with certainty.

By continuing the application of heat, the vapor will be raised in temperature and distilled from the surface of the oil at a sufficient rate to maintain combustion indefinitely. The temperature of the oil at this point is called the fire test or the burning point. When this temperature is reached, the thermometer must be removed and the flame be smothered with the lid *L*, which is a part of the testing equipment. Emergent-stem correction must be made for the fire test or burning point in the same way as for the flash point, if the thermometer is not the one provided for the apparatus by the manufacturer, for which stem correction has been made when making the graduations.

Oil-testing Machines.—The type of machine (Fig. 311) generally used in America, for testing an oil to determine coefficient of friction and effect in preventing undue heating in a bearing, consists of a horizontal steel shaft supported on two bearings, with one end overhanging its bearing.¹ A pendulum is suspended from the overhanging end by a bronze bearing. This bearing is made in two halves *B*, *B* so as to be removable for cleaning and scraping. In order to get results at all comparable on different machines, the bearing surfaces must be smooth and fit on the shaft as well as they can possibly be made. The pendulum is so arranged by means of an adjusting screw *N* at the bottom that any pressure is transmitted in its full amount to both halves of the bearing. The adjusting screw transmits its pressure on the lower half of the bearing

¹ For a detailed description of slightly different type (Golden's) see A. L. Westcott in the *Jour. A.S.M.E.*, July, 1913, pp. 1143-1167.

by an intermediary spring R , thus permitting closer adjustments. A pointer on the spring indicates on a scale attached to the pendulum the total pressure and also the total unit pressure, in pounds per square inch, on the two halves of the bearing. A thermometer inserted through a hole in the top of the pendulum, with its bulb resting on the shaft, indicates the temperature at the bearing surfaces. An iron ball is provided on some of these devices to counterbalance partially the weight of the pendulum and thus increase its sensitiveness. The deviation of the pendulum from the vertical is obviously proportional to the friction. This deflection is measured by a pointer moving over a circular arc. If the instrument is in proper adjustment, the reading on this scale divided by the total pressure exerted on the two halves of the bearing is the value of the coefficient of friction.

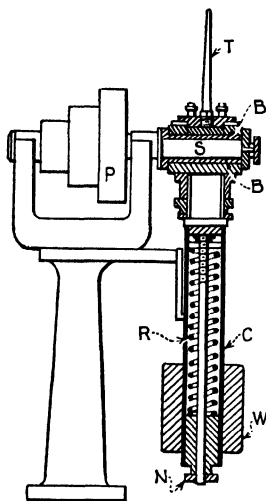


FIG. 311.

Sometimes when the apparatus is to be used at higher pressures than the normal spring permits, the spring can be replaced by

a stiffer one, made of heavier wire.¹

As the mathematical demonstration is usually stated:

P = total pressure on the two halves of the journal, in pounds.

p = total unit pressure per square inch of projected area on the two halves, in pounds per square inch.

T = tension in spring, in pounds.

W = gross weight of pendulum, in pounds.

R = effective arm of pendulum, in inches.

r = radius of journal, in inches.

α = angle of deflection of pendulum from the vertical.

F = total force of friction, in pounds.

f = coefficient of friction.

l = length of bearing surface.

Since each half is loaded, the total equivalent projected area is $2 \times 2rl$.

Total load on journal is $P = 2T + W$, and

$$p = \frac{P}{4rl} = \frac{2T + W}{4rl}.$$

¹ The maximum load a spring will carry is proportional to the cube of the diameter of the wire. When a new spring of heavier wire is made, the outside diameter of the coil must be made the same as of the normal spring, and the unstressed normal length must be the same.

Moment of friction is equal to moment of external forces or, since

$$f = \frac{F}{P},$$

$$Fr = Pfr = (2T + W)fr = W'R \sin a$$

in the case of the friction deflecting the **unbalanced** weight W' of the pendulum through the angle a . Then

$$f = \frac{W'R \sin a}{rP}.$$

In any machine of the type described, the term $\frac{W'R}{r}$ is a constant, and the scale on the arc indicating deflections on the pendulum is usually graduated to indicate values of $\frac{W'R \sin a}{r}$.

As a rule, the total and unit pressures stamped on the scale of the pendulum type of apparatus are not even nearly correct and it becomes necessary to calibrate the spring in the tester, if absolute values are to be calculated. This calibration is accomplished most easily by removing the brasses from the yoke, taking off the lower part of the pendulum which has the thread for tightening up on the spring, setting up the pendulum with the yoke downward in a testing machine for compression, applying suitable increasing loads on the spring, and measuring its deflection at each point. A stiff bar must be put through the yoke to take the pressure from the spring. This bar can be readily supported on iron blocks so that the pendulum will be free to move up and down with the varying compression on the spring. If all the supporting parts are rigid, the deflections of the spring can be measured by the distance between the head and the base of the testing machine.

Steam-engine Lubricators.—The proper oiling of an engine is most important. The operation of nearly all types of lubricating devices is easily understood, particularly when operated by the **gravity of the oil** or by a pump. Another type of lubricator for cylinder lubrication which is operated by the weight of a column of **condensed steam** is shown in Fig. 312. The pipe C , in which the condensed steam accumulates, must be made at least 2 feet long to give a sufficient head or pressure to feed the oil. The oil reservoir is in the cylindrical vessel below the condenser pipe. This is filled with oil through an opening in the top as the water which has accumulated in the apparatus is drained through the cock D . Water from the condenser pipe C is carried down to the bottom of the oil reservoir by the pipe shown in dotted lines on the left-hand side. Oil, being lighter than the water, remains in the upper part of the reservoir and is forced down through the pipe on the right-hand side and through the needle valve I by which the flow of oil can be regulated so that as small an amount as one drop in 2 or 3 minutes

passes into the steam pipe at *H* to mix with the steam going to the engine cylinder. Through the gage glass *S* the number of drops passing through can be observed. Another gage glass *L* on the side of the reservoir shows the relative amounts of oil and water. **When an engine is not operating, all oil cups and lubricators should be carefully closed.** The feed of oil from this lubricator is stopped by closing the valves *V* and *F*. A valve is usually provided on the bulb *B*, which should be closed when draining from *D*, so that the water in the condenser pipe *C*

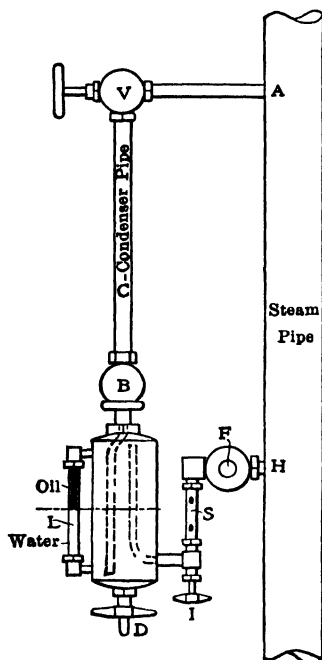


FIG. 312.—Engine cylinder lubricator operated by pressure of column of condensed steam.

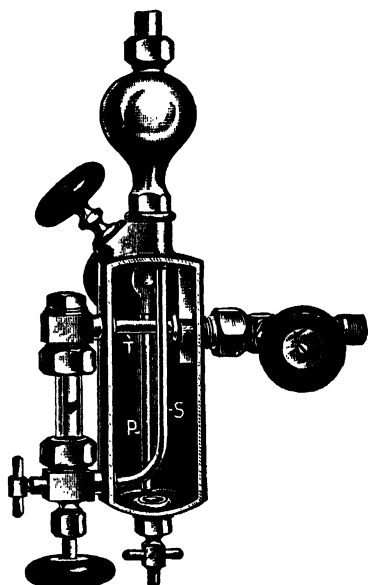


FIG. 313.—Cylinder lubricator similar to Fig. 312.

will not be lost and thus prevent the operation of the apparatus until a sufficient amount of condensed steam has accumulated to produce the pressure necessary to force the oil into the steam pipe.

A slight modification of the lubricator described is illustrated in Fig. 313. The condensed steam is brought to the bottom of the reservoir through the pipe *P*, which is open at its lower end. On the other hand, the pipe *S* is open at the top and the oil is forced by the pressure due to the head of water in the pipe above (not shown) through the gage glass on the left-hand side, then through the horizontal pipe *T* to which is attached the valve and nipple connected to the steam pipe supplying the engine. In this figure at the top of the reservoir a plug is shown, which is to be removed for filling with oil.

CHAPTER XXIII

TESTS OF HYDRAULIC MACHINERY

Tests of Boiler Feed Pumps.—In general engineering practice there are two classes of pumps commonly used for “feeding” the water to steam boilers. These are:

1. Motor- or belt-driven pumps.
2. Steam pumps.

Motor-driven feed pumps are quite generally used in steam power plants. The feed pumps are usually operated by direct connection to an electric motor or by belting.

Figure 314 shows a good modern example of a belt-driven pump. It has three plungers operated from a single shaft with the cranks set 120 degrees apart. A pump of this kind with three water cylinders is called “triplex.” One with two water cylinders (usually horizontal and steam driven) is called “duplex.” A “simplex” pump is one with a single water cylinder. The valves are accessible for cleaning or for repairs by removing the plates or “covers” *C, C, C*. The suction pipe is marked *S* and the discharge pipe *D*. An air chamber *A* is provided to produce, by cushioning air in it, a somewhat more steady flow than would otherwise be secured.

A **relief valve** should be provided on the discharge pipe to act as a safety valve in case the pressure in the line should get so high that the pump itself might be broken.

The power delivered to a belt-driven pump can usually be conveniently measured with some type of transmission dynamometer (see pages 172 to 180), while if it is direct-connected to a motor the efficiency of the motor may be obtained by disconnecting it, attaching a Prony brake to the shaft, and measuring the input to the motor with suitable electrical instruments (page 378).

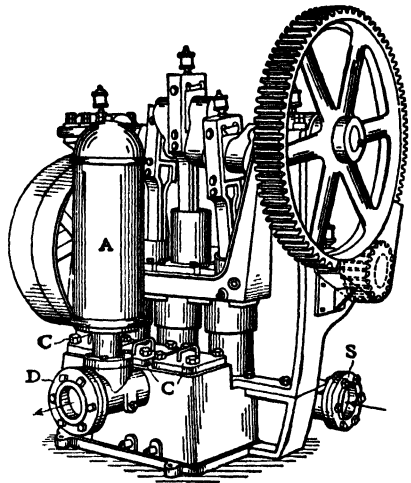


FIG. 314.—Belt-driven feed pump.

The equivalent work done "on the water" by the pump is found by multiplying the **total head**¹ (suction + discharge) in feet by the weight of water lifted (foot-pounds).

The quantity of water delivered can be determined by weighing or by calculating the flow over a weir or from an orifice (see pages 212 to 216).

Slip is the difference between the volume swept through by the plunger of the pump, or, in general, the piston displacement, and the actual volume of the water pumped at the required head.

In piston pumps direct-connected to the steam cylinder without a crank, the length of the stroke is usually variable, and some special method must be adopted for such tests to determine the average length of the stroke (page 551).

For a short test utilizing a transmission dynamometer belted to the pump for measuring the power, the following form may be used:

TABLE XLIII.—TEST OF A BELT-DRIVEN PUMP

Dynamometer Method

Type of pump.....	Made by.....
1. Duration of test.....	hr.
2. Diameter of plungers.....	in.
3. Length of stroke.....	ft.
4. Size of suction pipe.....	in.
5. Size of discharge pipe.....	in.
6. Speed of dynamometer.....	r.p.m.
7. Speed of pump.....	r.p.m.
8. Dynamometer reading.....	
9. Discharge pressure, in pounds per square inch (gauge).....	
10. Suction pressure, in pounds per square inch (gauge) or inches vacuum.....	
11. Temperature of water.....	°F.
12. Delivery head, in feet of water.....	
13. Suction head, in feet of water.....	
14. Total head in feet of water.....	
15. Net weight of water pumped per minute, in pounds.....	
16. Work done by pump, in foot-pounds per minute (Item 14 × Item 15).....	
17. Cubic feet water pumped per minute.....	

¹ If the **discharge** head is measured by a pressure gage on the discharge pipe, then the equivalent pressure in pounds per square inch corresponding to the difference in level between the surface of the water supply and the center of the gage must be added to get the **total head**. (One foot-head of water at about 62°F. is equivalent to 0.434 pound per square inch; and, conversely, 1 pound per square inch is equivalent to a head of 2.31 feet of water at the foregoing temperature.)

This is usually stated as the vertical distance between the two gages. A vacuum gage, however, on the suction pipe of a pump indicates the vacuum at the level of the *top of the suction pipe*, and not up to the center of the gage. This was shown by Professor Cooley by attaching a gage by means of a suitable fitting to the suction pipe of a pump so that the gage could be revolved above and below the pipe. It was observed that the reading of the gage remained constant which showed that in a suction pipe of a pump the water does not rise higher than the top of the pipe.

18. Plunger displacement, cubic feet per minute.....
19. Slip, in per cent [(Item 18 - Item 17) ÷ Item 18].....
20. Net work delivered to pump (by dynamometer), in foot-pounds per minute.....
21. Dynamometer horsepower (Item 20 ÷ 33,000).....
22. Pump horsepower (Item 16 ÷ 33,000).....
23. Mechanical efficiency (Item 22 ÷ Item 21).....
24. Capacity of pump, gallons delivered per 24 hr.....

A **direct-acting steam feed pump** like the one shown in section in Fig. 315 will be tested in a somewhat different manner, and a different set of observations is required.

In none of the so-called direct-acting steam pumps has a rotary motion been developed by means of which an eccentric can be made to operate the valve. It is, therefore, necessary to reverse the piston by an impulse derived from itself at the end of each stroke. This cannot be effected in an ordinary single-valve engine, as the valve would be moved only

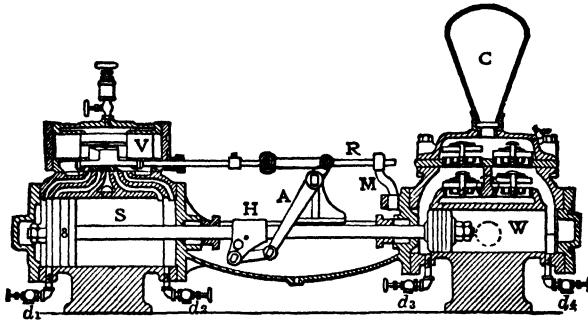


Fig. 315.—Direct-acting steam pump.

to the center of its motion, and then the whole machine would stop. To overcome this difficulty, a small steam piston is provided to move the main valve of the engine.

In these pumps the lever *A*, which is carried by the piston rod, comes in contact with the tappet when near the end of its motion and by means of the valve rod *R* moves the small slide valve which operates the supplemental piston. The supplemental piston, carrying with it the main valve *V*, is thus driven over by steam, and the engine is reversed. If, however, the supplemental piston fails accidentally to be moved, or to be moved with sufficient promptness by steam, the lug on the valve rod engages with it and compels its motion by power derived from the main engine.

Outside-packed steam pumps of the plunger type (Fig. 316) are now very commonly used for supplying boiler feed water, chiefly because at the pump end the only part subjected to ordinary wear is the packing of the plunger stuffing boxes. Steam is admitted at *A* and exhausts at *E*. The suction pipe is at *S* and the discharge pipe at *D*.

Suitable fittings for the attachment of indicators should be provided at both the steam and the water cylinders. If the pump is of the ordi-

nary direct-connected type, without a flywheel, like the one shown in Fig. 315, some provision must be made to make regular observations of the **length** of the stroke, as it is scarcely ever constant. One method is to attach a suitable arm to the cross-head *H* (Fig. 315) with a pencil at the end. Strips of tough paper can then be pasted on a board in such a position that the pencil will trace the lengths of the strokes. By shifting the position of the board every minute or two, records will be obtained from which the average length of the stroke can be estimated with considerable accuracy.

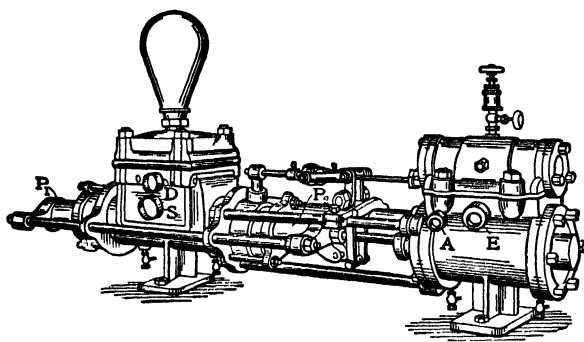


FIG. 316.—Outside-packed plunger feed pump.

Duty of a pump is usually defined as the number of foot-pounds of work "delivered" by the pump per 1,000,000 B.t.u. supplied. The heat units supplied by the engine are calculated by the A.S.M.E. Rules as the product of the weight of feed water used by the boiler and the total heat of steam at boiler pressure "reckoned from the temperature of the feed water." The total heat is to be corrected, of course, for moisture or superheat.

TABLE XLIV.—TEST OF DIRECT-ACTING STEAM-OPERATED PUMP (SINGLE WATER CYLINDER)

Type of pump.....	Made by.....
1. Duration of test.....	hr.
2. Diameter of steam cylinder.....	in.
3. Diameter of water cylinder.....	in.
4. Diameter of suction pipe.....	in.
5. Diameter of discharge pipe.....	in.
6. Diameter of rods.....	in.
7. Total number of strokes.....	
8. Single strokes per min.....	
9. Average length of stroke.....	ft.
10. Piston speed.....	ft. per min.
11. Steam pressure, gage.....	lb. per sq. in.
12. Moisture or superheat in steam.....	per cent or °F.
13. Total condensate.....	lb.
14. Total dry steam.....	lb.

15. Discharge pressure, gage.....	lb. per sq. in.
16. Discharge head.....	ft.
17. Suction head.....	ft.
18. Total head.....	ft.
19. Temperature of water.....	°F.
20. Weight of water pumped per min.....	lb.
21. Volume of water pumped per min.....	cu. ft.
22. Plunger displacement per min.....	cu. ft.
23. Slip.....	per cent
24. Horsepower steam cylinder, head end.....	i.hp.
25. Horsepower steam cylinder, water end.....	i.hp.
26. Horsepower steam cylinder, total.....	hp.
27. Water horsepower (useful work).....	hp.
28. Mechanical efficiency.....	per cent
29. Dry steam per indicated horsepower-hour.....	lb.
30. Dry steam per water horsepower-hour.....	lb.
31. Duty per 1,000 lb. steam.....	ft.-lb.
32. Duty per 1,000,000 B.t.u.....	ft.-lb.
33. Capacity of pump per 24 hr.....	gal.

Cooley Stroke-measuring Device.¹—Another method giving still greater accuracy is a counting device designed by Professor Cooley,

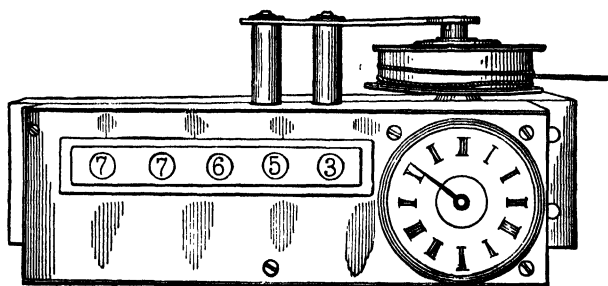


FIG. 317.—Cooley stroke-measuring device.

operated by a mechanism similar to that in a clock (Fig. 317). A cord from the instrument is attached to the cross-head of the pump and the clock mechanism moved by this cord integrates or sums the lengths of all the strokes. This apparatus has been developed for measuring accurately the length of the stroke of the type of pumps in which steam and water cylinders are direct-connected on the same piston rod, such, for example, as the ordinary steam feed pumps. In such pumps it scarcely ever happens that there are two strokes in succession that are of the same length and more or less approximate methods are usually adopted for obtaining the average length of the stroke during a test. With the stroke-measuring device referred to above, each individual stroke is accurately measured and is added by a counting device to the sum of all the other strokes that have preceded. As this apparatus is used in a test of a variable-stroke pump, the reading of the counter to the nearest

¹ Made by Engineering Shops, Ann Arbor, Mich.

inch can be recorded at the usual times for observations. The difference between two readings is the total length of all the strokes for the interval between observations. If, then, this difference is divided

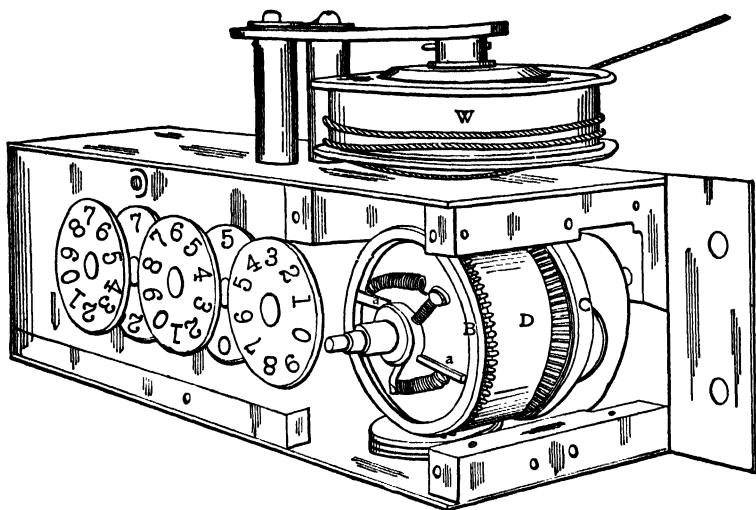


FIG. 318.—Mechanism of stroke-measuring device.

by the total number of strokes for the same time, the average length of the stroke can be determined accurately. An assembled view of this device is shown in Fig. 317 and its mechanism is shown in Fig. 318, which is an application of silent ratchet clutches.

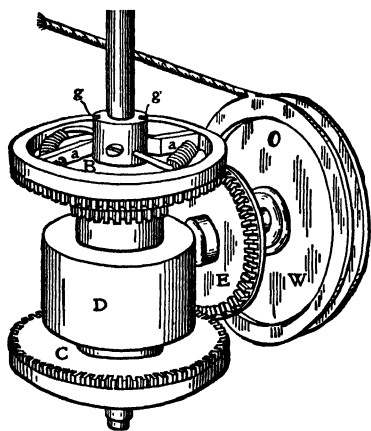


FIG. 318a.—Ratchet-clutch device of stroke-measuring integrator.

The apparatus is driven by the cord on the wheel *W*, which moves the ratchet wheels *B* and *C*. The operating mechanism, shown in Fig. 318a, consists essentially of two miter wheels *B* and *C*, meshing with another miter wheel *E* to which is attached the pulley *W*, carrying the cord that is fastened to the cross-head of the pump. At the top of the wheel *B* and at the bottom of *C* are silent ratchet clutches *a, a*, each of which operates in only one direction to grip the collars (top and bottom) in the notches *g, g*. The collars are securely attached to the central shaft by a set-screw. Only one of the collars is shown in the figure. For example, in

the operation of this mechanism the collar on the central spindle is gripped by the ratchets *a, a* in the miter wheel *B* during the forward stroke of the pump and are released during the backward stroke. On the other hand,

the ratchets on the miter wheel *C* grip the bottom collar on the central spindle during the backward stroke and release it on the forward stroke. In this way the movement of the cross-head of the pump constantly moves the central spindle of the instrument in the same direction. The miter wheels *B* and *C* can move freely on the central spindle and transmit motion to it only by means of the ratchets and the notches *g, g*.

Numbers on the horizontal plate (Fig. 317) are feet and those on the circular dial are inches.

A.S.M.E. TEST CODE (1926) FOR RECIPROCATING STEAM-DRIVEN DISPLACEMENT PUMPS (ABRIDGED)

The Code for Reciprocating Steam-driven Displacement Pumps applies to tests for determining the performance of the pump and engine, including reheaters, heaters, and jackets, if any, and jacket pumps, circulating pumps, condensate pumps, and vacuum pumps, which are concerned in their operation.

NOTE.—If any of the auxiliaries are independently driven, their efficiency may be determined separately from the main unit. For tests of pumping machinery and boiler plant combined, reference should be made to the Code for Complete Steam Power Plants (page 402).

Measurements.—The fundamental measurements for a duty trial of a reciprocating steam-pumping engine are: (a) amount of water pumped, in pounds; (b) average total head, in feet; (c) amount of steam supplied, in pounds; or the amount of heat supplied, in B.t.u.

The standard units for volume, head, and power are:

a. The standard unit for volume shall be the U. S. gallon or the cubic foot. The gallon unit shall be expressed in million gallons per day (m.g.d.) of 24 hours. The cubic foot unit, if used, shall be expressed in cubic feet per second. The standard U. S. gallon contains 231 cubic inches.

b. For temperatures not exceeding 85°F. the specific gravity of water may be taken as unity. (At 85°F. the specific gravity equals 0.9957.)

c. 7.48 gallons is equal to 1 cubic foot, and 448.8 gallons per minute is equal to 1 cubic foot a second; 694.4 gallons per minute is equal to 1,000,000 gallons per day of 24 hours.

d. The unit for measuring head shall be the foot. The relation between pounds per square inch and feet-head is 2.31 feet = 1 pound per square inch when the specific gravity of the liquid is unity.

e. In reading gages calibrated in pounds per square inch when other than water is handled, it is necessary to divide the reading in pounds by the specific gravity and multiply by 2.31. Example: 100 pounds per square inch pressure on brine of 1.25 gravity = 185 feet head, or the same pressure on oil of 0.85 gravity = 272 feet.

Instruments and apparatus required for a performance test of a steam-driven reciprocating displacement pump include: (a) tanks and platform scales for weighing water; (b) graduated scales attached to the water glasses of the boiler, if the feed water is measured; (c) pressure gages, vacuum gages, and thermometers provided with suitable wells; (d) steam calorimeter; (e) barometer; (f) revolution counter or other accurate speed-measuring device; (g) indicators; (h) a planimeter; (i) a dead-weight gage tester.

The gage on the discharge main should be attached near the pump discharge nozzle, and that on the suction main near the suction nozzle. These gages should be attached at an angle of 90 degrees to the direction of flow. The gage pipes should be provided with valves at the gages and petcock air vents.

When a suction or discharge surface condenser is in use as part of the unit under test, the water-head readings should be made outside and beyond the condenser so that the loss of head through the surface condenser will be charged against the main unit.

Leakage Test of Pump.—The tightness of the suction and discharge valves (and the plunger packing, if the pump is of the inside-packed construction) should be ascertained by applying the specified discharge pressure on them and observing the leakage by removing a hand-hole or manhole cover on the suction side of the part being tested. If essential to the object in view, the valves should be made tight; the plunger packing should leak only enough to insure that the friction of the packing on the plunger is not excessive. In direct-acting pumps with inside-packed plungers the amount of leakage may be estimated by closing the discharge valve and running the pump just fast enough to maintain the normal discharge pressure. If sufficient leakage is found to affect the capacity materially, the conditions should be corrected before proceeding with the test work.

Measurement of Water Pumped.—The quantity of water pumped should be determined by computing the actual plunger displacement in U. S. gallons or cubic feet. It is advisable, where apparatus is available, to check the total pumpage by the use of venturi tubes, Pitot meter, weir, calibrated orifice, etc., or by means of the use of a reservoir. In direct-acting pumps, the actual stroke of each plunger must be determined by the use of graduated stroke scales.

Starting and Stopping.—The engine and appurtenances should be first thoroughly heated and run under the prescribed conditions until uniformity is secured. When surface-condenser measurement is used, the duration of the test should not be less than 8 hours. Where the steam consumption is determined by measuring the feed water to the boiler, the duration of the test should be not less than 10 hours.

When a surface condenser is used, the test should start by commencing to weigh or measure the condensate and any other quantities of steam consumption involved, at the same time beginning the observations and other necessary test work. At the end of the allotted time the test is stopped by discontinuing the measurements and observations. When feed-water measurements are employed, the test should be started by carefully observing the steam pressure and water level in the boiler, and the level in the feed tank if measuring tanks are used, at the same time beginning the water measurements and taking up the routine work of the test. Toward the end of the prescribed time, when the water levels, steam pressure, and conditions of fire and draft are as near as practicable to the same points as at the start, the observations should be discontinued. If there are differences in the water levels, corrections must be applied to the water measurements.

Instruments should be read at intervals not greater than 15 minutes, and, if complete test data are required, indicator cards should be taken

from each end of each cylinder once each hour when the conditions are uniform, and oftener when there is much variation.

Heat Supplied.—The heat supplied to a steam pumping engine and its auxiliaries is the total heat content of the steam entering the engine and its auxiliaries, less the total heat returnable from the engine and its auxiliaries to boiler. The heat returnable is the sum of the total heat returned from the various sources as follows: (a) the heat returned in the condensate from the air pump, that is, the product of the heat of the liquid per pound, above 32°F., corresponding to its temperature; and the total weight of such condensate; (b) the heat returned in any feed-water heater using exhaust steam or steam from the working charge in the engine; (c) the heat returned in high-temperature jacket and other drain water.

Head.—The total head is determined from two elements, namely, the “discharge head,” which is the pressure in pounds per square inch expressed in feet-head in the discharge main referred to the center line of the pump cylinders (or other datum line as may be determined on), and the “suction lift” or “suction head,” which is the vacuum or head in the suction main referred to the same datum line. When the suction main is under a vacuum the head is found by adding the “suction lift” to the “discharge head,” both expressed in feet. When the suction main is under pressure the head is found by subtracting the “suction head” from the “discharge head,” both expressed in feet.

Discharge Head.—The discharge head is found by adding to the head in feet shown by the gage connected to the discharge main the vertical distance in feet between the center of the gage (if of the Bourdon spring type) or the lower surface of the mercury (if of the mercury type) above the center line of the pump cylinders (or other datum determined on). Should the gage be located below the datum line, this distance is to be subtracted instead of added.

Suction Lift.—The suction lift is found by adding to the reading in feet shown by the gage connected to the suction main the vertical distance in feet between the datum line noted and the point where the gage pipe connects to the main. Should the point of connection of the gage be located above the datum line, this distance is to be subtracted instead of added.

Suction Head.—The suction head, which obtains in cases where the suction pipe is under pressure, is found by subtracting the vertical distance in feet between the datum line and the center of the gage on the suction main (if of the Bourdon spring type) or the lower surface of the mercury (if of the mercury type) from the pressure in feet shown by the suction gage.

Capacity.—The capacity by displacement, in cubic feet per 24 hours, is found by multiplying the net or effective area in square feet of all the plungers by the length of the stroke in feet, the number of discharge strokes per minute, and the constant 1,440. The equivalent capacity in U. S. gallons per 24 hours is found by multiplying the capacity in cubic feet as determined above by the constant 7.48.

Water Horsepower.—The water horsepower is found by multiplying the weight of water pumped, in pounds per hour, by the average total head in feet, and dividing the product by 1,980,000.

Duty per 1,000 Pounds of Steam.—The duty in foot-pounds of work done per 1,000 pounds of steam is found by multiplying the number of pounds of water pumped during the test by the average total head in feet, dividing the product by the number of pounds of steam supplied during the test, and multiplying the quotient by 1,000.

Duty per 1,000,000 B.t.u.—The duty in foot-pounds of work done per 1,000,000 B.t.u. is found by multiplying the number of pounds of water pumped during the test by the average total head in feet, dividing the product by the number of B.t.u. supplied (as above), and multiplying the quotient by 1,000,000.

TABLE XLV.—TEST OF RECIPROCATING STEAM-ENGINE DRIVEN DISPLACEMENT PUMP
General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test; (7) duration of test.

Description and General Dimensions

8. Type of unit.....	
9. Number of steam cylinders.....	
10. Diameter of steam cylinders	{ h.p..... i.p..... l.p..... in. in. in.
11. Diameter of steam piston rods.....	in.
12. Stroke of steam pistons.....	in.
13. Plungers, number.....	
14. Single or double acting.....	
15. Inside or outside packed.....	
16. Diameter of plungers.....	in.
17. Stroke of plungers.....	in.
18. Diameter of plunger rods (if any).....	in.
19. Net area of each plunger.....	sq. ft.
20. Type of condenser.....	
21. Location of condenser.....	
22. Cooling surface in condenser.....	sq. ft.
23. Type and size of condenser pumps.....	
24. Type and size of exhaust pipe or receiver, feed-water heaters, steam reheaters, and jacket pumps or other auxiliaries a part of the unit.....	

Test Data and Results

Pressures

25. Steam pressure at throttle (gage).....	lb. per sq. in.
26. First receiver pressure (gage).....	lb. per sq. in.
27. Second receiver pressure (gage).....	lb. per sq. in.
28. H.p. jacket pressure (gage).....	lb. per sq. in.
29. I.p. jacket pressure (gage).....	lb. per sq. in.
30. L.p. jacket pressure (gage).....	lb. per sq. in.
31. Barometer.....in. of mercury, as observed at...°F. or...lb. per sq. in. abs.	
32. Corresponding absolute pressure (Items 25, 26, 27, 28, 29, and 30).....	lb. per sq. in.
33. Vacuum in exhaust pipe near l.p. cylinder...in. of mercury at...°F. lb. per sq. in.	
34. Corresponding absolute pressure near l.p. cylinder.....	lb. per sq. in.
35. Vacuum in condenser.....in. of mercury at...°F.....	lb. per sq. in.
36. Corresponding absolute pressure in condenser.....	lb. per sq. in.

Temperatures

37. Steam at throttle valve.....	°F.
38. Exhaust steam.....	°F.
39. Water pumped.....	°F.
40. Condensate leaving surface condenser.....	°F.
41. Condensate or feed water entering feed-water heaters.....	°F.
42. Condensate or feed water leaving feed-water heaters.....	°F.
43. Temperature rise in feed-water heaters.....	°F.
44. Condensate from jackets.....	°F.
45. Condensate leaving first receiver.....	°F.

46. Condensate leaving second receiver	°F
47. Condensate from other drains	°F.
48. Engine-room air	°F.
49. External air	°F.

Total Head

50. Discharge pressure by gage	lb. per sq in.
51. Equivalent discharge head	ft. of water
52. Vacuum or pressure shown by gage on suction main	ft. of water
53. Discharge head referred to datum line	ft. of water
54. Suction head or suction vacuum referred to datum line	ft. of water
55. Total head pumped against	ft. of water
56. Actual suction lift (if any) measured to highest point of discharge valve deck, (surface in which valve seats are inserted)	ft. of water

Quality of Steam near Throttle

57. Dryness factor of steam	per cent
58. Superheat in steam (if any)	°F.

Total Steam Quantities

59. Total steam as measured supplied to engine and the auxiliaries concerned in its operation as specified	lb.
60. Correction factor conforming to conditions agreed upon	per cent
61. Total "estimated steam" supplied conforming to conditions agreed upon	lb.

Total Pump Quantities

62. Total quantity of water pumped by plunger displacement ..	U. S. gal. or cu. ft.
63. Equivalent total weight of water pumped	lb.
64. Total quantity of water pumped as shown by other means of measurement	U. S. gal. or cu. ft.

Hourly Steam Quantities

65. Steam, as measured, supplied per hour	lb.
66. "Estimated steam" supplied per hour	lb.

Hourly Pump Quantities

67. Quantity of water pumped per hour by plunger displacement ..	U. S. gal. or cu. ft.
68. Equivalent weight of water pumped per hour	lb.
69. Quantity of water pumped per hour as shown by other means of measurement	U. S. gal. or cu. ft.
70. Slip in percentage of plunger displacement	per cent

Heat Supplied (based on steam supplied, as measured, Item 59)

71. Total heat per pound of steam at throttle	B.t.u.
72. Total heat in steam supplied	B.t.u.
73. Total heat returnable from engine to boiler	B.t.u.
74. Net heat supplied	B.t.u.
75. Net total heat supplied per hour	B.t.u.

Heat Supplied (based on total "estimated steam," Item 61)

76. Total heat per pound of steam at throttle	B.t.u.
77. Total heat in steam supplied	B.t.u.
78. Total heat returnable from engine to boiler	B.t.u.

79. Net heat supplied.....B.t.u.
 80. Net total heat supplied per hour.....B.t.u.

Speed

81. Total number of revolutions.....
 82. Total number of single strokes.....
 83. Actual length of stroke, if direct-acting.....ft.
 84. Revolutions per minute.....r.p.m.
 85. Single strokes per minute.....strokes per min.
 86. Piston and plunger speed per minute.....ft. per min.

Power

87. Total work done during test.....ft.-lb.
 88. Water horsepower.....water hp.
 89. Combined hydraulic and mechanical efficiency.....per cent
 90. Indicated steam horsepower.....i.hp.
 91. Mean effective pressure (referred to l.p. cylinder if multiple-expansion engine).....lb. per sq. in.
 92. Indicated water horsepower.....i. water hp.

Economy Results (based on steam supplied as measured, Item 65)

93. Steam supplied per indicated horsepower-hour.....lb.
 94. Heat supplied per indicated horsepower-hour.....B.t.u.
 95. Heat supplied per water horsepower-hour.....B.t.u.

Economy Results (based on "estimated steam,"* Item 66)

96. Steam supplied per indicated horsepower-hour.....lb.
 97. Heat supplied per indicated horsepower-hour.....B.t.u.
 98. Heat supplied per water horsepower-hour.....B.t.u.

Efficiency Results

99. Thermal efficiency (referred to indicated horsepower) based on steam as measured (2,545† ÷ Item 94).....per cent
 100. Thermal efficiency (referred to indicated horsepower) based on "estimated steam"* (2,545 ÷ Item 97).....per cent

Duty

101. Duty per 1,000 lb. of steam as measured.....ft.-lb.
 102. Duty per 1,000 lb. of "estimated steam"*.....ft.-lb.
 103. Duty per 1,000,000 B.t.u. based on steam supplied, as measured.....ft.-lb.
 104. Duty per 1,000,000 B.t.u. based on "estimated steam".....ft.-lb.

In cases in which the pump is driven by an electric motor, the characteristics and efficiency of the motor, and the power input to the motor should be obtained and used in accordance with the rules of the American Institute of Electrical Engineers.

Tests of Centrifugal Pumps.—Tests of pumps operating against low heads such as single-stage centrifugal pumps are made in the same

* The correction factor used shall be stated in percentage, and the basis and method of determining it described.

† The heat equivalent of one horsepower hour is 33,000 foot pounds per minute $\times 60 \div 778$ (foot pounds in 1 B.t.u.) = 2,545 B.t.u. per hour.

way as explained for the triplex belt-driven feed pump. It is desired, of course, from the results of the tests to compare the power supplied to the pump with the work done in lifting the water. Power supplied would probably be again measured by some form of transmission dynamometer, and the work done is calculated from the weight of water delivered and the total head against which the pump delivers.

Centrifugal pumps are frequently driven by direct-connected steam turbines. The horsepower required to drive the pump is then determined from a speed-power curve of the turbine similar to Fig. 272, page 375. It is obtained usually from a Prony brake test of the turbine. If the pump is driven by a variable-speed electric motor, a speed-power curve of the motor can be used. Usually, however, when a constant-speed motor is used it is simpler to determine an efficiency curve of the motor for varying power.

A.S.M.E. TEST CODE (1925) FOR CENTRIFUGAL AND ROTARY PUMPS (ABRIDGED)

This Code for testing centrifugal and rotary pumps applies to tests for determining the performance of the pump only and does not apply to the performance of piping, driving apparatus, or other auxiliaries.

Instruments required for measuring the head of water and speed of pump are (a) pressure gage; (b) vacuum gage; (c) mercury column; (d) revolution counter; (e) tachometer.

Operating Conditions.—The three elements constituting operating conditions are (a) the quantity of water delivered by the pump; (b) the total dynamic head; (c) the speed in revolutions per minute.

The prevailing total dynamic head under operating conditions may or may not be the correct designed head. If possible, this head should be brought to the designed value by suitable methods applicable to the local conditions of installation. Throttling by means of a valve in the discharge line is the most feasible means to raise the total head in case the operating head is less than the designed value, but there shall be no throttling of a valve in the suction line close to the pump.

Providing additional delivery openings in the discharge mains at a lower elevation will usually allow the total head to be reduced to the designed value. (Example: Opening hydrants in a system of street mains.)

Each of the various measurements for a given run shall be computed by averaging a series of instantaneous readings taken simultaneously at equal time intervals. Notations shall be made of extreme fluctuations at any instant. A sufficient number of readings shall be taken in making an observation so that the addition or elimination of a single reading

representing a maximum swing of the instrument shall not affect the average by more than 1 per cent.

The formulas given in this Code are *based on the use of water having a specific gravity of unity*. Unless otherwise stipulated, the temperature of the water during the test shall not exceed 85°F.

Duration of Test.—The duration of the test depends on the method of driving the pump and on the method used to measure the discharge. Since guaranteed conditions generally cover only one point in the head, speed, and capacity characteristics, it is recommended that data for additional points on these curves be taken when special information is desired. In these additional tests the capacity should be varied in steps covering a range of at least 10 to 15 per cent above and below the normal tested delivery of the pump and curves plotted of the results. This procedure gives complete pump characteristics and furnishes a check on the separate tests in the smoothness of the two separate curves (head—capacity and shaft horsepower—capacity).

Standard Units.—The standard unit for volume shall be the U. S. gallon or the cubic foot. The gallon unit shall be expressed in gallons per minute (g.p.m.) or million gallons per day of 24 hours (m.g.d.). The cubic foot unit shall be expressed in cubic feet per second. The standard U. S. gallon contains 231 cubic inches.

For temperatures not exceeding 85°F. the specific gravity of water may be taken as unity. (At 85° the specific gravity is 0.9957.) Seven and forty-eight hundredths gallons is equal to 1 cubic foot and 448.8 gallons per minute is equal to 1 cubic foot a second; 694.4 gallons per minute is equal to 1,000,000 gallons per day of 24 hours.

The unit for measuring head shall be the foot. The relation between a pressure expressed in pounds per square inch and that expressed in feet of head is 1 pound per square inch = 2.31 feet (water) when the specific gravity of the liquid is unity. (One inch of mercury = 1.132 feet of water.) In reading gages calibrated in pounds per square inch, when other than water is handled, or when the temperature of the water is such that the weight per cubic foot is greater or less than 62.35 pounds, it is necessary to divide the reading in pounds per square inch by the specific gravity and to multiply by 2.31. Example: 100 pounds per square inch pressure on brine of 1.25 gravity is equivalent to 185-foot head; the same pressure on oil of 0.85 gravity is equivalent to 272-foot head; and a like pressure of 100 pounds per square inch on water at 210°F. having a gravity of 0.96 is equivalent to 240-foot head.

The unit for measuring power shall be the horsepower delivered to the pump shaft and designated as shaft horsepower (s.hp.).

Measurement of Quantity of Water.—Methods of water measurement are divided into primary and secondary methods, according to whether the method is in itself an absolute measurement of quantity or merely the measurement of an effect of quantity of rate of flow. Of these methods the only primary methods are those of measurement by volume or weight.

These are measurements by weir, venturi meter, nozzle, or Pitot tube. Practically all these instruments involve a coefficient whose value and range of application are really well-known, but if not they should be determined by a separate test.

Measurement of Head.—The *net total head* is the *total dynamic head* produced by the pump and is represented by the difference between the absolute heads at the discharge and suction nozzles, any difference between the elevations of these nozzles with respect to a fixed datum and any difference between the velocity heads at the points of measurement being taken into account.

With the suction and discharge velocities equal and the suction and discharge gages at the same elevation, the *true dynamic head* is the difference in absolute pressure that these gages register.

Where accuracy is desired, it is recommended that mercury manometers be used instead of Bourdon gages when the head against which the pump is operating is 50 feet or less. When Bourdon gages are used, it is recommended that drain cocks be placed immediately below the gages and that frequent tests be made to determine that the pipe connections of the gage are full of water. Mercury gages shall be calibrated under the actual conditions of temperature, specific gravity, etc., prevailing in the test by comparing the gage reading with the height of a water column. With any form of gage care shall be taken to eliminate any leaks in the connecting piping, even if small, and to avoid the trapping of air in the connecting pipe or hose.

The **total dynamic head** H may be expressed by means of a formula.

Where h_d = discharge-gage reading, in feet, corrected to datum elevation.

h_a = atmospheric-pressure head, in feet, corresponding to the barometer.

$H_d = h_d + h_a$ = absolute discharge pressure head, in feet.

h_s = suction-gage reading, in feet, corrected to datum elevation.

$H_s = h_s + h_a$ = absolute suction pressure head, in feet.

V_d = mean discharge velocity, in feet per second.

V_s = mean suction velocity, in feet per second.

$$H = \left(\text{Head equivalent to total energy of discharge} \right) - \left(\text{Head equivalent to total energy of suction} \right)$$

$$H = \left(H_d + \frac{V_d^2}{2g} \right) - \left(H_s + \frac{V_s^2}{2g} \right)$$

$$= H_d - H_s + \frac{V_d^2 - V_s^2}{2g}$$

With Bourdon gages see Figs. 319 and 320.

If mercury U-tubes are used, the procedure is the same (see Figs. 321 to 322).

For differential gages (Bourdon type) the gage reads exact value of $H_d = H_s$ [see Fig. 323 (1)].

If a mercury U-tube¹ is used as a differential, care must be exercised that all the connections are either entirely filled with water or entirely filled with air [see Fig. 324 (2) and 325 (1)].

Measurement of Power Input.—Measurement of power input to the pump, that is, the shaft horsepower (s.h.p.) of the pump, falls into two general classes. Some measurements are those which in themselves determine the actual power or torque delivered to the pump and are therefore made entirely during the test, with some form of transmission dynamometer. Other measurements are those which involve measurement, during the pump test, of power input to the driving element and the previous or subsequent determination of the relation of the power input to the power output of this driving element under the identical conditions of the pump test (efficiency or calibration of the driving element).

Water Horsepower.—The water horsepower (w.h.p.) is found by the following formulas:

$$\text{w.h.p.} = \frac{Q \times W \times \text{total head in feet}}{550},$$

¹ Mercury U-tubes to have uniform bore at working range h_d or h_s . To check this, observe deflection above and below zero point. If unlike, add or subtract half the difference to f and f_1 . Add when lower-leg reading is greatest, subtract when upper-leg reading is greatest.

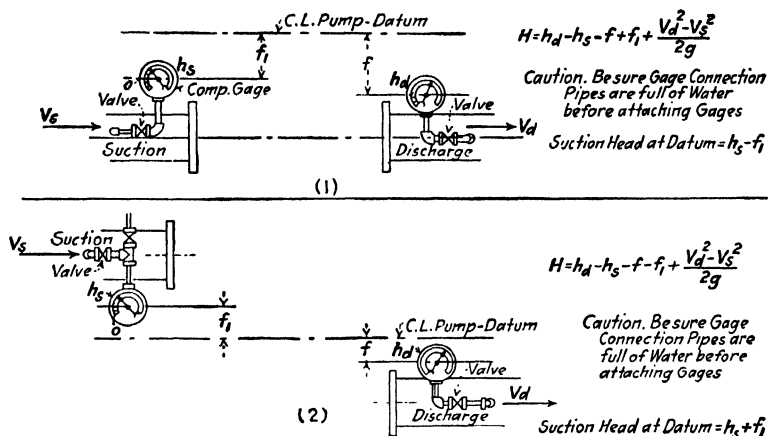


FIG. 319.—Suction head above atmospheric pressure (by Bourdon gages).

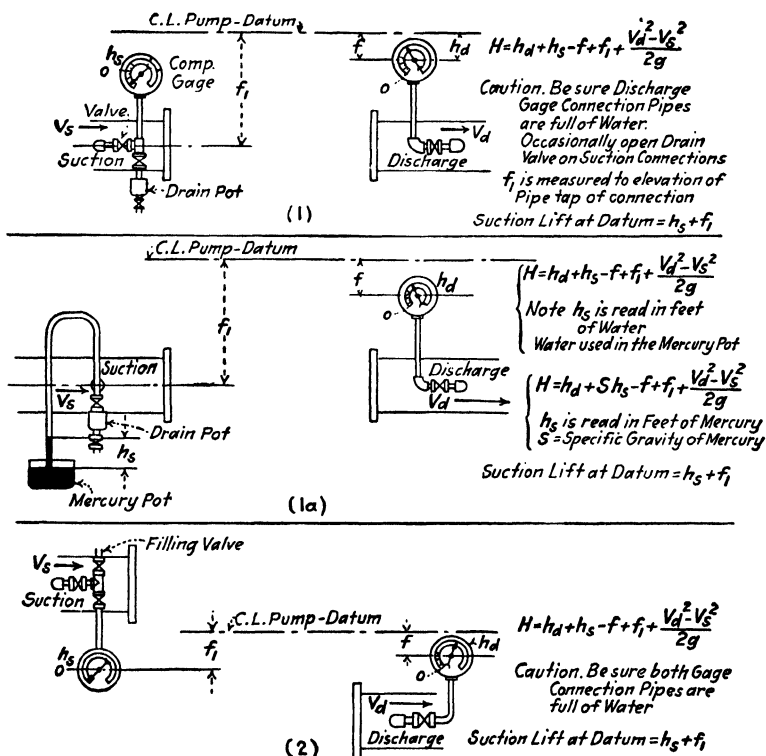


FIG. 320.—Suction head below atmospheric pressure.

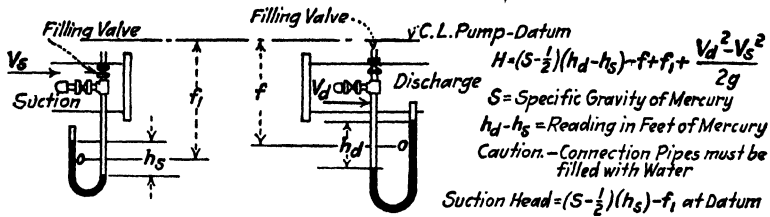


FIG. 321.—Suction head above atmospheric pressure (by U-tubes).

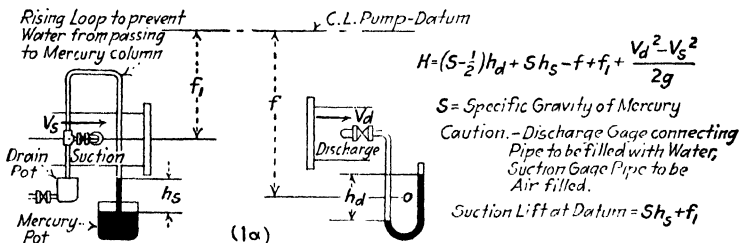
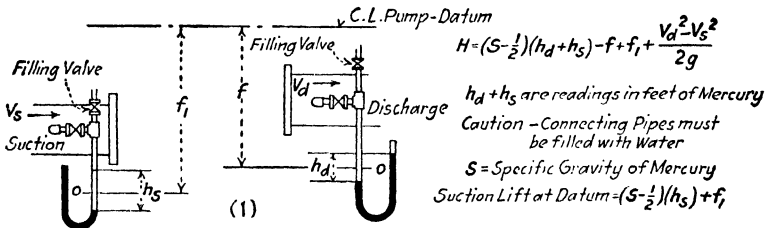


FIG. 322.—Suction heads less than atmospheric pressure (by manometers).

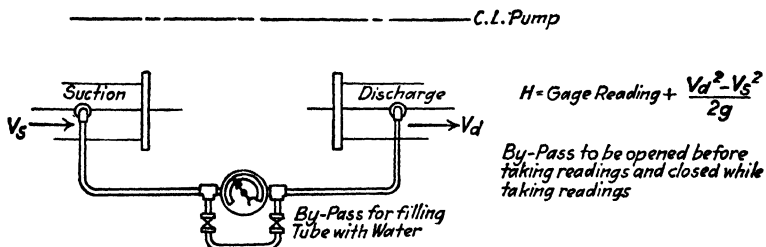


FIG. 323.—Heads of water measured by differential gages (Bourdon type).

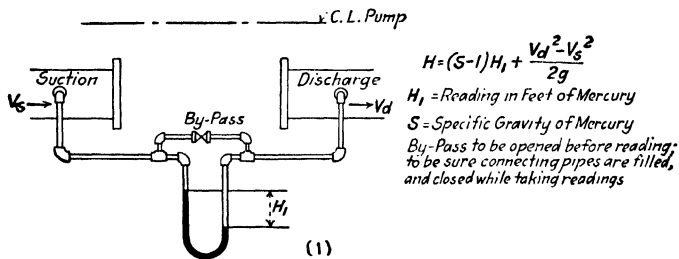
where Q = number of cubic feet of liquid pumped per second.

W = weight of liquid, in pounds per cubic foot.

When the weight of water is 62.32 pounds per cubic foot—density at standard temperature of 68°F.—

$$\text{w.hp.} = \frac{\text{g.p.m.} \times \text{total head in feet}^*}{33,000 \times 0.120}$$

Suction Head either above or below Atmospheric, Discharge Head above Atmospheric



Suction Head and Discharge Head Below Atmospheric

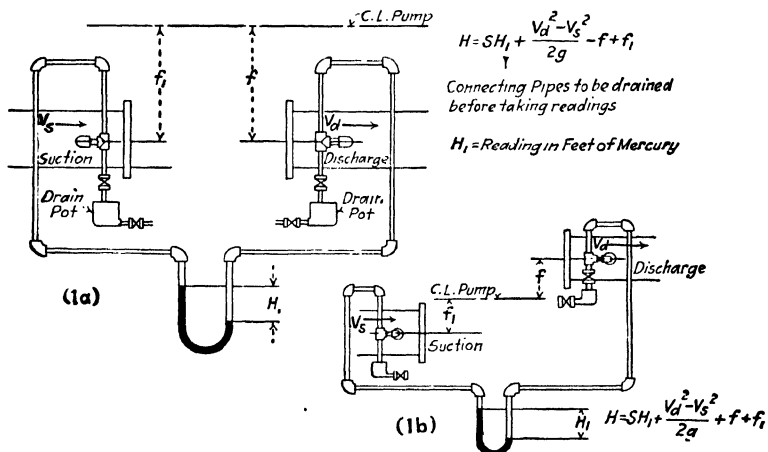


FIG. 324.—Heads of water measured by "differential" manometers.

Efficiency.—The efficiency of a pump is the ratio of the energy converted into useful work to the energy supplied to the pump, that is, the ratio of the water-horsepower (w.hp.) output to the shaft-horsepower (s.hp.) input, or

$$\text{Efficiency in per cent} = \frac{\text{w.hp.}}{\text{s.hp.}} \times 100.$$

* Where g.p.m. is gallons per minute, and 0.120 is a conversion factor with which to divide gallons of water per min. (at 68°F.) to obtain the equivalent weight in pounds, thus,

$$\frac{\text{G.p.m.} \times 231}{1,728} \times 62.32 = 0.120 \text{ pound,}$$

the numbers 231 and 1728 being, respectively, the cubic inches in a gallon and in a cubic foot, and 62.32 the weight of a cubic foot of water.

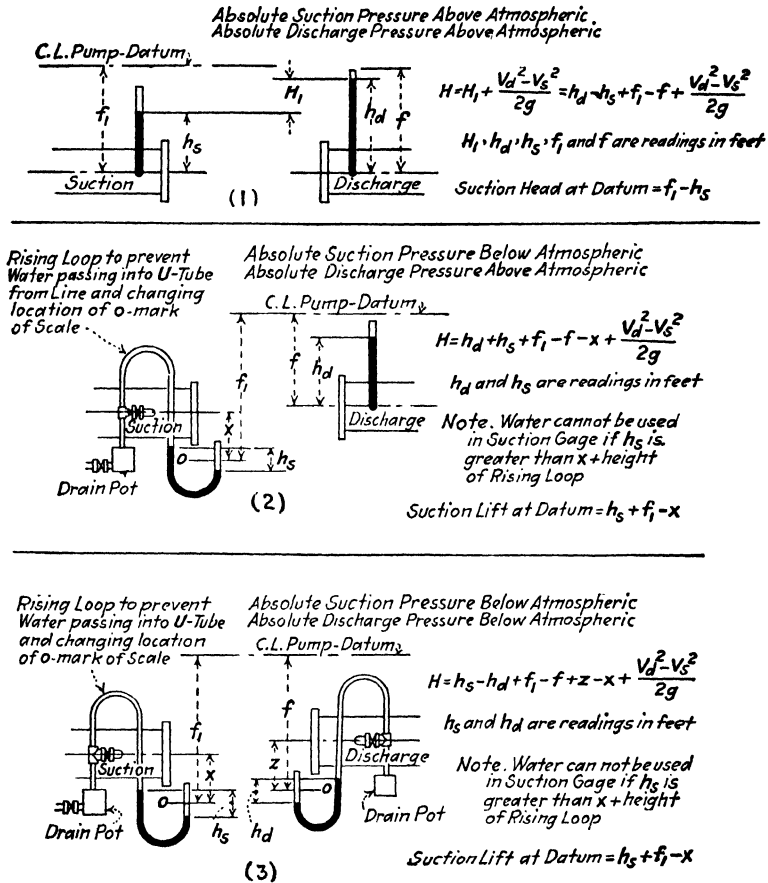


Fig. 325.—Heads of water measured by water gages and U-tubes.

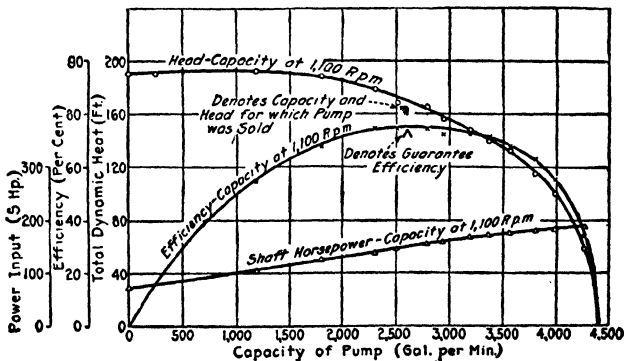


Fig. 326.—Characteristic curves of centrifugal pump.

When it is impossible to obtain the designed speed on test owing to variation of frequency in the electric current or other causes, *corrections in the capacity, head,*

and shaft horsepower to correspond to the designed speed may be made from test data. These corrections are based on the fact that within small differences in speed the capacities vary directly as the speeds, the heads as the squares of the speeds, and the shaft horsepowers as the cubes of the speeds where the speed variations do not exceed 3 per cent.

It is customary to present the results of a test in the form of a single sheet upon which are drawn three curves expressing respectively the relation between (a) efficiency and capacity, (b) head and capacity, and (c) shaft horsepower and capacity (see Fig. 326). The values used in plotting these curves are those computed for a given definite speed of the pump, usually that at which the pump operates at best efficiency or that at which it was designed to operate.

If the pump under test is of the rotary displacement type, such as the *gear pump*, its *volumetric displacement corresponding to the speed* developed in the test may be computed, and the difference between this displacement and the quantity actually pumped may be stated as the *slip*, expressed as a percentage of the displacement.

TABLE XLVI.—TEST OF CENTRIFUGAL OR ROTARY PUMPS

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test; (7) builder.

Pump Data

9. Type (centrifugal or rotary, horizontal or vertical, shaft, split, or solid casing, enclosed or open-type impeller, single or multistage, spiral casing, with or without diffusion vanes, etc.).....
10. Rated capacity.....
11. Rated head.....ft.
12. Rated speed.....r.p.m.
13. Liquid pumped (clear, muddy or sandy water, or other fluid, specific gravity, temperature, etc.).....
14. Method employed for measuring quantity of water.....
15. Method employed for measuring power.....
16. Method employed for measuring head.....
17. Duration of period of measuring quantity of water.....hr.
18. Duration of period of measuring power.....hr.
19. Duration of period of measuring head.....hr.
20. Size of discharge outlet.....in.
21. Size of suction inlet.....in.
22. Time in service and general condition of pump.....

Driver Data

23. Type (motor, turbine, gear, belt, etc.).....
24. Name of builder.....
25. Builder's type and serial number.....
26. Rated conditions (horsepower, speed, electrical or steam conditions, etc.).....
27. Time in service.....

Results

28. Volume of water pumped.....cu. ft. or.....g.p.m.
29. Discharge head absolute:

- (a) Computed velocity in suction pipe at gage.....ft. per sec.
- (b) Computed velocity in discharge pipe at gage!.....ft. per sec.
- (c) Head due to difference in velocity heads, plus or minus.....ft.
- 30. Suction head, absolute.....ft.
- 31. Total head on pump(p. 548).....ft.
- 32. Speed.....r.p.m.
- 33. Water horsepower.....hp.
- 34. Horsepower input to driver.....hp.
- 35. Efficiency of driver.....per cent
- 36. Shaft horsepower or horsepower input to pump.....hp.
- 37. Pump efficiency.....per cent
- 38. Plot (see Fig. 326).....
- 39. Test conducted by.....
- 40. Specific speed of pump = $\text{r.p.m.} \times \sqrt{Q} \div H^{\frac{1}{2}}$ at maximum efficiency point
(from characteristic curve, Fig. 326). In this equation Q is cubic feet of liquid
pumped per *second*, and H is the total dynamic head (p. 561), in feet.

Tests of Impulse Water Wheels.—Impulse wheels used to operate with water under pressure consist usually of a series of buckets attached

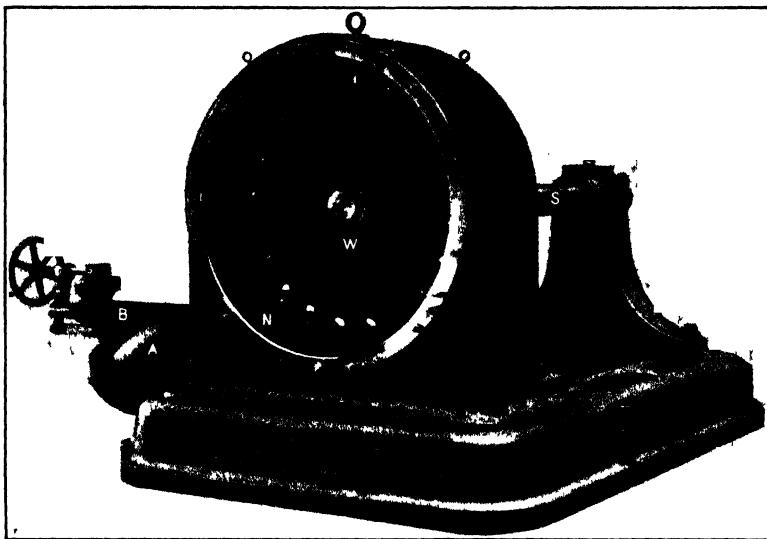


FIG. 327.—Typical impulse water wheel.

to the periphery of a disk or wheel. Figure 327 shows a typical impulse wheel. It is, therefore, impracticable to measure the head directly in feet but it is done usually by measuring the pressure with a gage near the nozzle *N*. When the center of the gage is at a higher level than the center of the nozzle discharging on the wheel, then this difference in level must be added to the head calculated from the gage pressure to determine the total head under which the wheel is operating. Power developed is measured usually by a Prony brake connected to the shaft *S*. In all tests where a large quantity of water is used, the temperature

of the water should be recorded and the weight corresponding should be used.

Laboratory tests for a *given head* are usually run when varying both the load and speed. Make the first test with the load on the Prony brake as light as possible consistent with fairly steady operation of the wheel, and then take a series of tests, increasing the load in increments to reduce the speed about 100 revolutions per minute in each succeeding test. Duration of test at each speed should be from 20 to 30 minutes with observations taken every 2 minutes. The following form may be used for tests:

TABLE XIV.7.—TEST OF IMPULSE WHEEL

General data:

1. Date.....	
2. Name of wheel and nominal horsepower.....	
3. Kind of bucket... ..	
4. Number of buckets.....	
5. Angle of buckets.....	
6. Diameter of bucket wheel, inches.....	
7. Area of nozzle and delivery pipe, square inches.....	
8. Coefficient of discharge for type of nozzle.....	
9. Diameter of brake wheel, inches.....	
10. Length of brake arm, inches.....	
11. Tare of brake, pounds.....	
12. Duration of test.....	
13. Average temperature of water, degrees Fahrenheit.....	
14. Average pressure by gage at wheel, pounds per square inch.....	
15. Average head at wheel in feet*.	
16. Quantity of water for total run in pounds.....	
17. Quantity of water in pounds per minute.....	
18. Cubic feet of water per minute.....	
19. Foot-pounds of work per minute calculated from Items 15 and 17.....	
20. Revolutions per minute.....	
21. Net weights on brake, pounds.....	
22. Horsepower as measured by brake.....	
23. Overall efficiency of motor, per cent Item 22 ÷ Item 19 × 33,000.....	

* Corrected for vertical distance from the center of the gage to the center of the nozzle.

Curves.—Plot a curve for each head with speed for abscissas and efficiency in per cent for ordinates, also curves for the ratio of the velocity of the periphery of the wheel v_p to the theoretical velocity due to the head v_t ; that is, $\frac{v_p}{v_t}$ for abscissas and the maximum horsepower developed for ordinates.¹

Air Lifts.—Pumping water by compressed air has had extensive application in recent years. There are several methods in use, but the most successful is that using air expansively for raising water as shown diagram-

¹ Plot curves showing effect of head on efficiency if several tests are run at different heads.

matically in Fig. 328, called the "air lift." It consists of a delivery pipe D set down into the well and a small pipe for compressed air having a nozzle N at the end and entering the discharge pipe as shown. It is a more usual construction to admit the air to the discharge pipe through holes from an annular chamber encircling it, but the method of operation is practically the same as shown. Water is raised by the buoyancy of the air. Let h_1 be the depth of submersion of the delivery pipe measured to the point where the air enters, and h_2 be the total lift measured from the same point. Pressure of air at the place of entrance must be theoretically equal to the pressure corresponding to the head of mixed water and air above it in D . This pressure decreases, however, as the air rises and expands so that at the top of D the pressure is little above atmospheric. Work required of the compressor varies with the difference between the depth of submersion of the delivery pipe h_1 compared with the net lift $h_2 - h_1$. The pressure required is then comparatively high and the efficiency low. On the other hand, a very small submergence necessitates a relatively large quantity of air to produce the required velocity, so that again the efficiency is low. Maximum efficiency, usually about 50 per cent, is obtained when the net head $h_2 - h_1$ is from 15 to 30 feet. At 150 feet the efficiency is scarcely ever as much as 20 per cent.

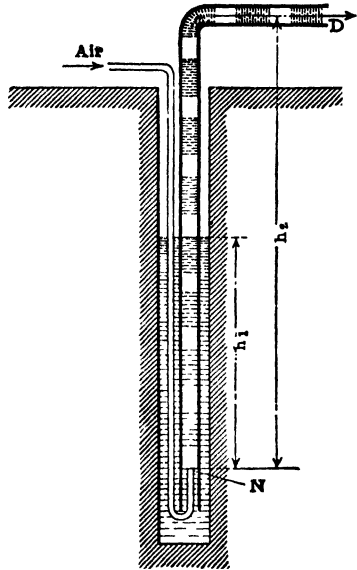


FIG. 328.—Air lift.

The following quantities should be determined in a test: (1) horsepower of air compressor, (2) volume of free air (see page 452 and 457) compressed, (3) weight of water pumped, (4) net lift ($h_2 - h_1$), (5) efficiency, which is the ratio of the work equivalent of lifting the water to the work done in compressing the air, both in foot-pound units.

Tests of Hydraulic Rams.—A section of a typical hydraulic ram is shown in Fig. 329. It consists of an air chamber H , to which is connected the discharge pipe I . There is a check valve G opening into the air chamber from the lower chamber A into which water is brought by the pipe S . There is a waste valve at B . This valve is weighted and opens inward. By means of a nut J on the stem of this valve the lift or amount of opening of the valve can be regulated. When water is supplied to the ram, it escapes through the waste valve B with a velocity corresponding approximately to that due to the head under which the water is supplied. The effect of this velocity head is to reduce the pressure on the upper side

of the valve so that it becomes unbalanced and closes suddenly. Then the momentum of the column of water in the pipe *S* becomes sufficient to open the valve *G* and will discharge some water into the discharge pipe *I* against a considerable head. As soon as the pressures become equalized, the valve *G* closes, the waste valve *B* opens, and water from the supply pipe is again "wasted." This alternate action is produced with regularity, and as a result the water in the supply pipe acquires a certain "backward and forward" wave motion. As the rule is generally stated, the length of the supply pipe leading from the reservoir to the ram must be at least five times the head. This length is necessary to secure some resistance to this "backward and forward" wave motion. A small air

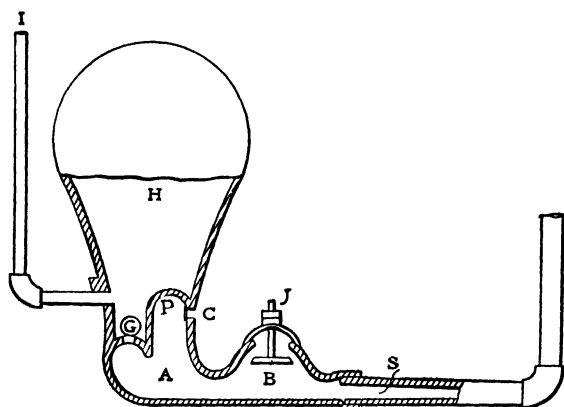


FIG. 329. —Section of a simple hydraulic ram.

chamber shown at *P*, with a check valve *C* opening inward to supply air, is provided in many of these rams, as it improves the efficiency. The rate of opening of the waste valve or the number of pulsations in a given time can be varied by changing the weight on its stem. This apparatus is tested usually by measuring the supply and the discharge heads, the weight of water discharged through the delivery pipe w_1 , and that passing through the waste valve w_2 in pounds per minute. Then the available energy in the water is $(w_1 + w_2)h_s$, where h_s is the supply head; and the useful work is w_1h_d , where h_d is the discharge head,¹ then,

$$\text{Efficiency} = \frac{w_1 h_d}{(w_1 + w_2) h_s}, \quad (107)$$

and the **capacity** *Q* in gallons per 24 hours is $Q = 1,440w_1q$, where *q* is the fraction of a gallon of water in a pound. Satisfactory runs of 20 minutes' duration can usually be made, each run being made with a different lift or "stroke" of the waste valve *B*. Observations of the

¹ Both the supply and the discharge heads must be measured, of course, from the same datum or "zero" level.

heads should be taken every 5 minutes if they are variable, and weighing as often as is necessary, depending on the size of the tanks used. The effect on the efficiency of increasing the lift or "stroke" of the waste valve from $\frac{1}{8}$ inch by increments of $\frac{1}{8}$ inch is interesting.

Curves.—Plot curves with length of "stroke" as abscissas and take for ordinates:

1. Efficiency.
2. Capacity in gallons per 24 hours.
3. Strokes per minute.

Figure 330 shows a slightly different form of ram, as made commercially. The principle of operation is, however, the same as the one in

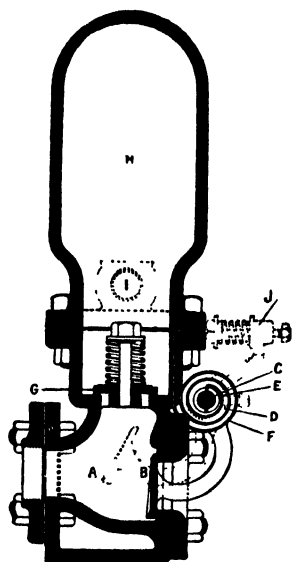


FIG. 330.—Commercial type of hydraulic ram.

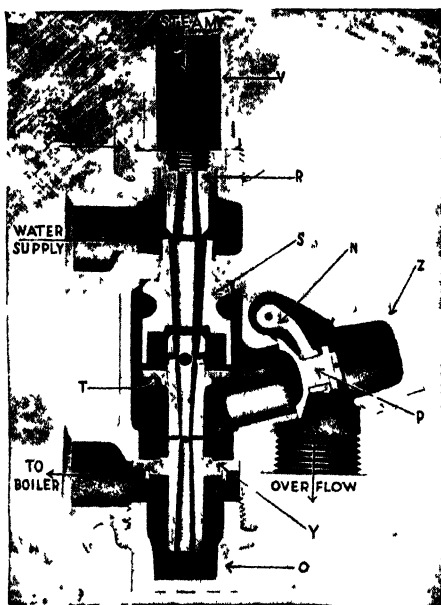


FIG. 331.—Single-tube steam injector.

Fig. 329. Letters used for marking the parts are the same in the two figures.

Tests of Injectors.—The injector is known particularly in stationary service as the device used for pumping water into the boiler when the feed pump fails. One of the various forms of injectors sold commercially is shown in Fig. 331. The **steam supply**, the **suction** or water supply, the **delivery** or discharge, and the **overflow** are marked clearly.

Method of Operating Injectors.—The method to be given, although applicable particularly to the one described, is, however, more or less generally applicable to all makes. Open **wide** both the **steam-** and **water-supply** (suction) valves. Then **close** the water-supply (suction)

valve slowly until the overflow ceases. Regulate the rate of delivery by closing the water-supply (suction) valve. Before testing an injector or indeed even before trying to operate a new injector, inspect the pipe fittings and particularly the valves on the water-supply pipe to observe whether they are tight. It is not at all unusual to find that the valve is not airtight, and for this reason it is a very good practice always to put some new wicking in the space for packing around the stem of the valve on the water-supply pipe; and turn up tightly the cap over the packing.

Method of Testing.—For the testing of injectors the arrangement of apparatus consists usually of two barrels supported on platform scales, or carefully calibrated tanks fitted with gage glasses. During a test the injector draws water from one barrel or tank and discharges it into the other. A test of an injector must be made, of course, with established conditions, that is, with a flying start. This may be accomplished by having the injector draw water from the supply tank but discharge water through a by-pass connection on the discharge pipe until the test is to begin. For this preliminary operation of the injector the level in the supply tank can be maintained very closely, at any point marked, by manipulating a “quick-opening” valve. When the test is to begin, close as quickly as possible this valve on the pipe discharging into the supply tank and turn the discharge from the by-pass into the delivery tank. To make this adjustment, all the valves to be operated should be of the “quick-opening” type. The pressure against which the injector is to operate is secured by **throttling** the discharge pipe by means of a globe or an angle valve placed between the injector and the by-pass on the discharge pipe. The quick-opening valve would not be satisfactory. The suction head is measured from the middle of the injector to the **average** level of the water in the supply tank. The discharge head is obtained by adding to the head in feet corresponding to the pressure indicated by the gage the distance in feet from the center of the gage to a horizontal line through the middle of the injector. The temperatures of the water in the supply and delivery pipes must be observed. The injector is stopped at the end of the test by closing the steam valve.

The following form for tests when the steam supplied is dry, saturated, or wet is similar to the one used at Purdue University. If the steam supplied is superheated, suitable corrections must be made in the data and in the formulas given. Tests should be made for several discharge pressures. Notes explaining the calculations required are given.

TABLE XLVIII.—TEST OF AN INJECTOR

Make of injector.....	Date.....
Number.....	
Size of connections: steam..... in. dia.; water..... in. dia.; discharge	
..... in. dia.; area of discharge (= a).....	sq. in.

Diameter (minimum) of lifting tube.....	in.; forcing tube.....	in.
a. Duration of test.....	hr.	Barometer..... in. mercury at..... °F.
b. Steam pressure (average) p_s		lb. per sq. in. abs.
c. Delivery pressure (average) p_2		lb. per sq. in. (gage)
d. Maximum pressure against which injector will discharge, p_{max}		lb. per sq. in. (gage)
e. Suction head (average), feet, h_1		
f. Delivery head (average), feet, h_2		
g. Temperature of supply (average) t_1		°F.
h. Temperature of delivery (average) t_2		°F.
i. Pounds water supplied per hour, w_w		
j. Pounds water and steam delivered per hour, w_m		
k. Cubic feet of water delivered per hour, Q		
l. Wet steam per hour, $w_s (= w_m - w_w)$		
m. Dry steam per hour, $w'_s (= xw_s$ where x is quality of steam).....		
n. Water delivered per pound wet steam, pounds ($= w_w \div w_s$).....		
o. Water delivered per pound dry steam, pounds ($= w_w \div w'_s$).....		
p. Velocity of discharge, feet per second, $v (= 144Q \div 3,600a)$		
q. Energy delivered, raising injection water, B.t.u. per hour.....		
r. Energy delivered, heating injection water, B.t.u. per hour.....		
s. Energy delivered, velocity of discharge, B.t.u. per hour.....		
t. Total energy delivered, B.t.u. per hour.....		
u. Energy supplied, B.t.u. per hour.....		
v. Thermal efficiency as a boiler-feed apparatus.....		per cent
w. Thermal efficiency as a pump.....		per cent
x. Horsepower.....		
y. Dry steam per horsepower per hour, pounds.....		
The energy of raising injection water $= [w_w(h_1 + h_2) + w_s h_2] \div 778$ B.t.u. per hour.		
The energy of heating injection water $= w_w(h_{f2} - h_{f1})$ where h_{f1} and h_{f2} are the heat of the liquid corresponding to t_1 and t_2 , B.t.u. per hour.		
The energy of discharge $= w_m v^2 \div (2g \times 778)$, B.t.u. per hour.		
The total energy delivered = Item q + Item r + Item s.		
The energy supplied $= w_s(xh_{f0s} + h_{fs} - h_{f2})$, where h_{f0s} and h_{fs} are the heat of evaporation and heat of liquid corresponding to p_s , and h_{f2} corresponds to t_2 .		
The thermal efficiency as a boiler-feed apparatus $= 100 \times \frac{\text{Item } t}{\text{Item } u}$.		
The thermal efficiency as a pump $= 100 \times \frac{\text{Item } q + \text{Item } s}{\text{Item } u}$.		
The horsepower $= \frac{w_w(h_1 + h_2) + w_s h_2}{60 \times 33,000}$.		
The dry steam per horsepower per hour $= w'_s \div \text{Item } x$.		
The pump duty $= \frac{1,000,000 + \text{item } p}{\text{Item } t}$		

$$= \frac{1,000,000[w_w(h_1 + h_2) + w_s h_2]}{778w_s(xr_s + q_s - q_2)} \quad (109)$$

The weight of steam found by direct weighing may be checked by calculating (assuming radiation loss negligible) a "heat balance" in which this weight will be the only unknown; thus for this condition,

$$w_s(xh_{f0s} + h_{fs} - h_{f2}) = \frac{1}{778} \left[w_w(h_1 + h_2) + w_s h_2 + (w_w + w_s) \frac{v^2}{2g} \right] + w_w(h_{f2} - h_{f1}),$$

or, approximately,

$$w_s = \frac{w_w \left[h_1 + h_2 + 778(h_{f2} - h_{f1}) + \frac{v^2}{2g} \right]}{778(xh_{f0s} + h_{fs} - h_{f2}) - h_2} \quad (110)$$

Curves.—Plot with discharge pressures for abscissas, curves with both thermal efficiency and capacity (gallons or pounds of water per 24 hours) as ordinates.

Tests of Pulsometers.—A type of steam pump called a pulsometer is illustrated in section in Fig. 332. In the form shown here it consists of two chambers *AA*, joined by tapering necks into which a ball *C* is fitted so as to move in the direction of the least pressure between seats in these

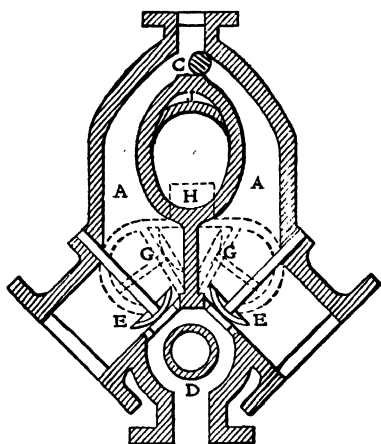


FIG. 332.—Steam pulsometer.

tapering passages. The chambers *AA*, on opposite sides, are connected by means of check or clack valves *EE* with the “induction” chamber *D*. Water is delivered through the passage *H*, which is connected to the chambers through the valves *G*. Between the chambers is also a vacuum chamber *J*, connecting them with the “induction” chamber *D*. Small air valves, moving inward, supply air to the chambers *AA* by opening when the pressure is less than atmospheric. Its operation is explained briefly as follows: Starting with the left-hand chamber full of water and with a vacuum in the right-hand chamber,

this latter chamber will fill with water, which by its momentum, due to its rushing in suddenly, pushes back the valve *C* toward the left. Now **during this time** steam has entered the left-hand chamber to the left of the valve *C* (before it has shifted) and by exerting a pressure on the surface of the water it forces it through the check valve *G*, first into the delivery passage *H* and then into the air chamber *J*. Then the steam in this left-hand chamber condenses in contact with the cold water and forms a vacuum, permitting the repetition of this cycle of events, except that the operations in the two chambers are reversed.

Since all the steam used is condensed and discharged with the water lifted, the analysis of the operations in a pulsometer are similar to those in the familiar types of injectors, except that the steam acts in the pulsometer by pressure instead of by impact as in the injector.

The method of testing a pulsometer is exactly the same as for an injector (page 571). Since the theory of operation of the pulsometer

and injector is the same, the calculated results and curves to be plotted are similar.

The A.S.M.E. Test Code for Hydraulic Power Plants is not included here for lack of space. It was published in *Mechanical Engineering*, April, 1922, and is now available in pamphlet form, as approved April, 1927.

CHAPTER XXIV

TESTING OF GOVERNORS

A.S.M.E. TEST CODE (1927) FOR SPEED-RESPONSIVE GOVERNORS

Tests for Speed Regulation—Regulations with Variation of Load.—

The regulating characteristics are best determined by operating the prime mover on a load by itself, such as a water rheostat, which may be thrown on and off at will, in whole or in part. The prime mover should be operated with a wide-open throttle, or gate, and under normal conditions as to temperature and pressure at the throttle and exhaust pressure. The speed should be recorded after various increments of load, commencing at zero and terminating with the maximum load the unit is intended to carry, with not less than eight intermediate increments of load. The load increments should be applied slowly. Before taking any readings, ample time should be allowed after changing the load to be sure that the speed is constant for the new steady load and the prime mover has settled down to constant conditions. At each observation the load and speed should be accurately determined. Inlet and outlet water heads in the case of hydraulic turbines, and throttle temperatures and pressures and exhaust pressures in the case of steam prime movers, should be recorded as evidence that normal operating conditions have been maintained. Other pressures, such as stage pressures with a steam turbine, are illuminating and should be recorded.

For this test, speeds may be taken by a mechanical counter and determined by counting the revolutions per minute over a period of not less than 5 minutes. An indicating tachometer should also be used as a check on speed constancy during the period of observation. The test with increasing load is to be repeated with decreasing load. The test is to be regarded as unsatisfactory if there is any easily observable hunting of the governor or surging of load while observations are being made, although a slight periodic speed variation, not in excess of $\frac{1}{4}$ per cent in the case of steam prime movers and $\frac{1}{2}$ per cent in the case of hydraulic turbines, is not regarded as objectionable.¹

The speed variation between the tested loads may be determined directly from these data. The results of the foregoing tests are to be plotted with load as the abscissas and speed as the ordinates. The curve will show the amount the governor system deviates from the straight-line characteristic. Speed regulation is generally expressed in terms of a ratio and with reference to two extremes of load. Thus the speed regulation between two loads is defined as the ratio of the difference between the extreme speeds and the arithmetical mean of the extreme speeds. Thus if $N_1 =$

¹ Regulation of hydraulic turbines is more difficult than that of steam prime movers because water is nearly incompressible, and the action of the governor is affected by eddies, whirls, etc.

revolutions per minute for load *A* (the higher speed) and N_2 = revolutions per minute for load *B* (the lower speed), then

$$\text{Speed regulation between loads } A \text{ and } B = \frac{N_1 - N_2}{\frac{N_1 + N_2}{2}} = \frac{N_1 - N_2}{N},$$

where N is the *arithmetical mean* (page 492) of the two extreme speeds. It is usual to express this ratio as a percentage.

It is seldom, particularly in the case of large electric generating units, that a unit may be operated alone with means for varying the load. Governor tests must therefore be carried out frequently with the prime mover operating in parallel with other units on commercial loads. The speed-regulation test can, however, be carried out if the system on which the prime mover is operating may have its speed or frequency varied by the amount of the probable governor variation. If such a variation cannot be permitted, then all thought of speed-regulation tests must be abandoned. The test may be carried out as follows:

1. Connect the unit whose performance is to be observed to the power system.
2. By means of the governor controls of the various units operating, bring the frequency of the system sufficiently above normal so that the total variation from normal frequency during the conduct of the test shall be a minimum. This can be accomplished by setting the frequency above normal by one-half the amount by which the frequency of the unit under test may be expected to vary between the two extremes of load under consideration.
3. While the frequency of the system is being adjusted in this way, set the governor control of the unit under test so that it carries friction load only.
4. When these preparations have been completed, load is slowly shifted in the desired increments to the unit under test by means of the governor controls of the other units which are operating. After the application of each increment of load, the load carried by the unit under test and the corresponding speed should be determined and recorded.

This test should then be conducted in the reverse direction, unloading by the desired decrements. Obviously, the governor adjustment of the unit under test should not be altered in any way during the test.

Regulation with Constant Load.—An approximate determination of the degree of departure from constant speed of the prime mover with constant load and operating conditions may be made by noting the extremes of speed variation on a sensitive tachometer connected to the prime mover. More accurate determinations may be made with a recording tachometer. For rapid variations, this test had best be made using an electric tachometer generator and *oscillograph* (see page 583).¹ The generator should be rigidly connected to the prime mover so that there can be an angular variation between it and the prime mover. Speed variations of low period under constant load and operating conditions are generally due to defects in the governing mechanism. In the case of hydraulic turbines, small variations of speed of low period frequently occur due to disturbed conditions of flow in the turbine draft tube, particularly at the lighter loads. Variations of higher frequency also occur in reciprocating engines during a single revolution, caused by uneven turning moments.

The test record shall state: (a) regulation from . . . kilowatt to . . . kilowatt . . . , in per cent; (b) maximum speed varying with constant operating condition and constant load of . . . kilowatt . . . per cent, which

$$= 100 \times \frac{N_{\max} - N_{\min}}{\frac{N_{\max} + N_{\min}}{2}}.$$

¹ The handling of the oscillograph during these test should be done by one skilled in its use.

Some indication concerning the regulation of a governing mechanism at constant load, also concerning the stability of the apparatus, may be secured by disturbing the governor when the prime mover is carrying certain loads by forcibly pulling the levers up or down and then immediately releasing them and recording the amount of movement of the valve (or gate) produced by this disturbance, and also recording the time required thereafter for the mechanism to return to a condition of stability.

The test record shall state: The prime mover carrying . . . kilowatts, and the regulating mechanism being disturbed by moving the levers up or down . . . inches, the movement of the control valve (or gate) amounted to . . . per cent of the total movement of the control valve (or gate), and the regulating mechanism returned to a condition of stability in . . . seconds.

Test for Sensitiveness.—This test is to be made by bringing about a slow change of load and observing the change in speed necessary to cause change of position of the complete regulating mechanism. This test shall be carried out by graphically recording the change of speed and valve position and by observing the operating conditions, which latter, except the change in load and speed, must be constant. The test is difficult to carry out with sufficient accuracy to show discrepancies with apparatus that operates well. It serves, however, to show mechanical defects in poor apparatus. A measure of sensitiveness may be made during the speed-regulation tests by noting the difference in speeds between ascending and descending loads. If any observations of sensitiveness are made in this manner, care must be taken that there be no reversal of progress of increasing or decreasing load; that is, in ascending loads the load must not be first raised and then lowered before readings are taken, and *vice versa*. The test record shall state: Change in speed necessary to cause change of position of the complete regulating mechanism . . . per cent.

Tests for Rapidity.—For this purpose a recording tachometer or oscillograph (see page 583) must be employed. The motion of the chart should be approximately from $\frac{1}{2}$ to 1 inch per second. This test will determine the instantaneous variation and the final variation for a given change of load, and the time consumed between the load being changed and the prime mover settling down to constant speed. Prime-mover specifications sometimes state that with certain prescribed instantaneous changes of load the maximum change of speed shall not exceed a specified amount, and that the prime mover will settle down to constant speed within a prescribed time limit. This test gives these data. Sample diagrams illustrating the governor action in the case of steam prime movers are shown in Fig. 333. Figure 334 shows the action in the case of hydraulic turbines.

One test may be made by means of a multiple-element oscillograph, which can be made to exhibit the time lag of all the involved functions, the respective oscillograph elements being arranged to portray the action of such of the following functions as desired, of which (a), (b) and (c) are imperative: (a) time of load change on prime mover; (b) speeds of prime mover; (c) time of movement of control valve or gate; (d) movement of speed-responsive device; (e) fluid pressures operating the relay mechanism; (f) movement of deflecting or by-passing mechanism (in hydraulic installations); (g) pressure changes in prime-mover supply in the case of hydraulic installations. The observations of (a), (b), and (c) are within the scope of this code. The remainder partake of the nature of governor analysis for the purpose of discovering detail defects in the mechanism. The oscillograph element indicating load may be standardized to read actual loads. This is not, however, necessary, as the principal function of the oscillograph is to record accurately the time of change.

The report of the test, besides the inclusion of one or the other of the foregoing diagrams in Figs. 333 and 334 will be a statement that with an instantaneous change of load from . . . to . . . , the maximum speed variation was . . . per cent of the

original, and that in . . . seconds the prime mover had settled down to a constant speed of . . . per cent of the original.

An approximate determination of the speed at which the relay mechanism is capable of operating the control valves may be made with the prime mover at rest and with the relay disconnected from the speed-responsive device by throwing the relay from one extreme of its position to the other and measuring with a stop watch or other timing means the time taken for the mechanism to have completed its travel.

In many cases governors are provided with means by which the speed of the prime mover may be varied some amount above and below normal, which is usually expressed as a percentage. The test may be made by merely observing the extremes of speed which may be attained with the means provided, maintaining the prime mover under the control of the governor. The governor tests should include statements of the following: (a) maximum speed at full load for which governor may be adjusted, . . . per cent above normal; (b) maximum speed at no load for which governor may be

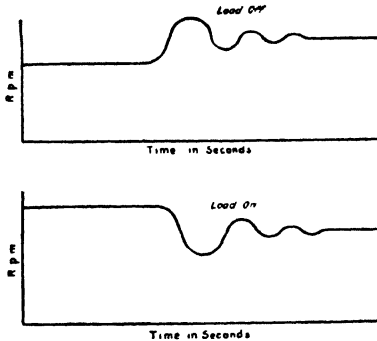


FIG. 333.—Oscillograph diagrams of governor on steam turbine.

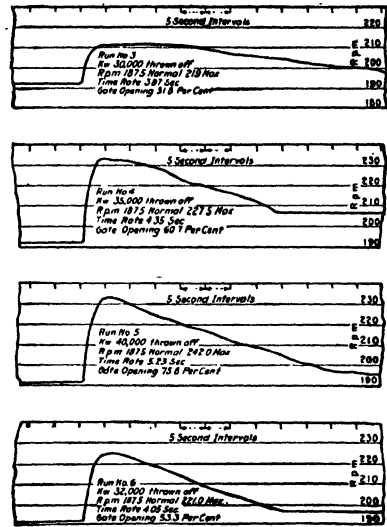


FIG. 334.—Oscillograph diagrams of governor on hydraulic turbine.

adjusted, . . . per cent above normal; (c) minimum speed at full load for which governor may be adjusted, . . . per cent below normal; (d) minimum speed at no load for which governor may be adjusted, . . . per cent below normal.

Tests of Governing Force—Governor Strength for 1 Per Cent Change in Speed.—This should be determined for the inner and the outer positions of the weights of the speed-responsive device, as in some designs the strength of the governor varies greatly between both of these positions. The governor strength for 1 per cent change of speed may be determined by actual test by operating the prime mover at no load and normal speed and adjusting the linkage between governor and valve gear so that the governor will be close to its high-speed position and the governor control adjusted so that the prime mover operates at $99\frac{1}{2}$ per cent of standard speed. Then apply a force to the governor sleeve and determine the additional force required to bring the speed of the prime mover to $100\frac{1}{2}$ per cent of standard, or 1 per cent higher speed. This can be readily accomplished by measuring the increase of force exerted by the adjustable governor control, as, for example, the auxiliary governor spring. This will be the governor force for 1 per cent change of speed in the high-speed position. This observation shall be followed by determining the position of the sleeve and then changing the adjustments of linkage between the governor and the valve gear until the speed of the prime mover is lowered to $99\frac{1}{2}$ per cent of standard speed. The

position of the governor sleeve is again recorded and the difference between the two positions corresponds to the governor travel for 1 per cent change in speed in the high-speed position.

The linkage between governor and valve gear should next be adjusted so that the governor will be close to low-speed position and the governor control adjusted so that the prime mover operates at $100\frac{1}{2}$ per cent of the standard speed. Then, as before, determine the force on the sleeve (now in the opposite direction) to bring the speed of the prime mover to $99\frac{1}{2}$ per cent of the standard value. This, as before, may be conveniently accomplished by measuring the reduction of tension in the control spring. This amount will be the governor force for 1 per cent change of speed in the inner position. With the prime mover operating as above and at $99\frac{1}{2}$ per cent speed, measure the position of the sleeve and change the adjustments of the linkage between the governor and the valve gear to bring the speed of the prime mover back to $100\frac{1}{2}$ per cent of the standard value, and again measure the position of the sleeve. The difference between the two governor positions corresponds to the governor travel for 1 per cent change in speed in the inner position. During these observations, care must be taken that the governor parts at no time come in contact with the governor stops.

In carrying out the foregoing test it may be found that for a 1 per cent change of speed the sleeve travel will be small and accuracy of measurement uncertain. If desired, the speed may be varied in the foregoing manner a greater percentage, and the result divided by that percentage to reduce it to a standard of a governor strength for 1 per cent change.

Work Capacity for 1 Per Cent Change in Speed.—This is the product of the arithmetical mean of the two governor forces for 1 per cent change of speed, and the arithmetical mean of the two sleeve travels for 1 per cent change of speed. The test record shall contain the statement: (a) governor force for 1 per cent change in speed in the inner position . . . lb.; (b) governor force for 1 per cent change in speed in the outer position . . . pounds; (c) mean work capacity of governor for 1 per cent change in speed . . . inch-pounds. The governor strength and work capacity may, if more convenient, be calculated by dividing the weights, supporting arms, and revolving masses into a series of weights and calculating the centrifugal moments of these on the sleeve, for both inner and outer positions.

The integration of the rotating masses as above described is unnecessary in governors where the center of gravity of the weights and arm is approximately in the center of the governor weight. The moment on the governor sleeve should be calculated for $99\frac{1}{2}$ and $100\frac{1}{2}$ per cent speed, the governor force being the difference between the two. The centrifugal force referred to the sleeve may, with advantage, be calculated for a number of positions of the governor travel and plotted on squared paper against the travel of the sleeve, when characteristics of the governor may be portrayed.

Strength of Relay Apparatus.—Where relay apparatus is interposed between the governor and control valves or gates, the force that it is capable of giving should be determined by trial. The test should first be by means of a steam-engine indicator, determining the average pressure on both sides of the operating piston, while the latter is being caused to travel at maximum speed for the full travel. For this test, with the prime mover at rest, the relay should be disconnected from the governor and the relay plungers quickly thrown from one position to the other. This gives a measure of the force available for motion at maximum speed, this force being the product of the average pressure difference and the area of the piston. The locked force of the relay, that is, the maximum power available for moving the valve gear or gates should they stick, is the difference between the fluid-actuating pressures on either side of the operating piston multiplied by the piston area.

The test record shall contain the statement: (a) force of relay when locked . . . pounds; (b) force of relay available for maximum speed of operation . . . pounds.

Work capacity of relay would be the foregoing forces multiplied by the travel of the piston. The test record shall contain the statement: (a) work capacity of the relay for slow motion . . . inch-pounds; (b) work capacity of the relay for maximum speed of operation . . . inch-pounds.

TABLE XLIX.—TEST OF SPEED-RESPONSIVE GOVERNOR

General Information

(1) Date of test; (2) location; (3) owner; (4) builder; (5) test conducted by; (6) object of test; (7) kind and size of prime mover; (8) type of governor (centrifugal, spring or weight, relay, gear or belt driven, etc.).

Speed regulation	Beginning of valve opening	No load	Partial loads, kw.	Maximum load, kw.	Extreme valve opening
9. Load.....					kw.
10. Load—per cent of normal.....					per cent
11. Time.....					min.
12. Atmospheric pressure.....					lb. per sq. in. abs.
13. Pressure at inlet (at throttle).....					lb. per sq. in. abs.
14. Pressure at various stages.....					lb. per sq. in. abs.
15. Temperature at inlet (at throttle).....					°F.
16. Temperature at various stages.....					°F.
17. Pressure at discharge (exhaust pressure).....					lb. per sq. in. abs.
18. Effective head (hydraulic turbine).....					ft.
19. Inlet valve or gate opening.....					per cent
20. R.p.m. increasing load— N_i					r.p.m.
21. R.p.m. decreasing load— N_d					r.p.m.
22. Mean of Items 20 and 21 = $\frac{N_i + N_d}{2} = N$					r.p.m.
23. Difference $N_i - N_d$					r.p.m.
24. Speed regulation $100 \frac{N_i - N_d}{\frac{N_i + N_d}{2}}$ (p. 577) from....kw. to kw.....					per cent
25. Sensitiveness: change of speed necessary to cause change of position of complete regulator mechanism (... kw. load).....					per cent
Stability					
26. Maximum speed variation with constant operating conditions and load					
$= 100 \times \frac{N_{\max} - N_{\min}}{\frac{N_{\max} + N_{\min}}{2}} \text{ with load of ...kw. (p. 577) per cent}$					
27–30. Time, in seconds .. for mechanism to settle to a condition of stability following a manual disturbance of in., up or down (measured at sleeve), producing per cent movement of the control valve or gate (p. 578), the prime mover carrying.....					kw.

Rapidity

31.	Load released {	from.....	kw.
32.		to.....	kw.

33. Speed before load is released.....r.p.m.
 34. Momentary maximum speed.....r.p.m.
 35. Speed after fluctuations have ceased.....r.p.m.
 36. Maximum increase in speed, $100 \times \frac{\text{Item 37} - \text{Item 36}}{\text{Item 36}}$ per cent
 37. Time of movement of control valve or gate.....sec.
 38. Time between release of load and prime mover settled down to constant speed (p. 578).....sec.
 39. Load imposed { from.....kw.
 40. { to.....kw.
 41. Speed before load is imposed.....r.p.m.
 42. Momentary minimum speed.....r.p.m.
 43. Speed after fluctuations have ceased.....r.p.m.
 44. Maximum decrease in speed, $100 \times \frac{\text{Item 44} - \text{Item 45}}{\text{Item 44}}$ per cent
 45. Time of movement of control valve or gate.....sec.
 46. Time between imposition of load and prime mover settled down to constant speed (p. 578).....sec.

Range of Speed Adjustment

47. Total stroke of governor sleeve.....in.
 48. Stroke of sleeve employed in regulating between no load and maximum load..in.
 49. Maximum speed at full load for which governor may be adjusted while operating (p. 578).....per cent above normal
 50. Maximum speed at no load for which governor may be adjusted while operating.....per cent above normal
 51. Minimum speed at full load for which governor may be adjusted while operating.....per cent below normal
 52. Minimum speed at no load for which governor may be adjusted while operating.....per cent below normal

Governor Force and Work Capacity

53. Governor force on sleeve for 1 per cent change in speed in inner or maximum load position.....lb.
 54. Governor force on sleeve for 1 per cent change in speed in outer or no-load position.....lb.
 55. Mean work capacity of governor for 1 per cent change in speed (p. 580—State whether determined by test or by calculation).....in.-lb.

Governor with Relay

56. Relay fluid pressure.....lb. per sq. in.
 57. Relay piston area.....sq. in.
 58. Force of relay when locked.....lb.
 59. Mean effective fluid pressure on relay during full stroke at maximum-speed motion.....lb. per sq. in.
 60. Mean force of relay available for maximum speed motion.....lb.
 61. Travel (employed) of relay piston.....in.
 62. Travel of regulating valve stem or gate.....in.
 63. Time of maximum speed motion.....sec.
 64. Work capacity of relay for slow motion (p. 581).....in.-lb.
 65. Work capacity of relay for maximum speed motion.....in.-lb.

The **oscillograph** is an instrument for observing (and photographing usually) alternating-current waves. The one shown in Fig. 335 is easily portable and is very well suited for use by schools for undergraduate work; by power companies for wave-form analysis and substation use; by electric railways for recording instantaneous values of current in subway trains, elevated lines, etc.; by small electrical-accessories manufacturers for studying ignition, starting-motor characteristics, etc.

The instrument includes an especially sensitive galvanometer of very small dimensions, a 4-volt incandescent lamp with straight-line filament,

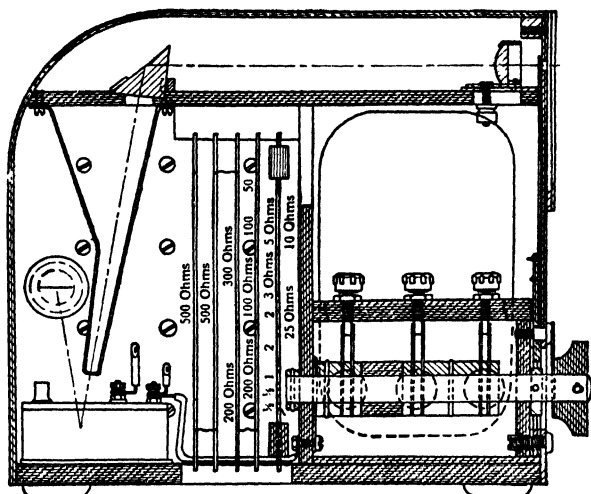


FIG. 335.—Oscillograph (Westinghouse).

optical system, vibrator resistances, and mechanism for throwing the one vibrator into six different circuits, inserting proper resistance, and shifting a cross-hair to mark each wave.

The "heart" of the instrument is the permanent-magnet galvanometer. The moving coil consists of two parallel conductors in a strong magnetic field, with the current to be measured passing in opposite directions in these conductors. As the magnetic field is at right angles to the conductors, one bends one way and the other the other way. A tiny mirror is cemented across the two conductors in the center of the air gap of the permanent magnet so that its angular twist is exactly proportional to the instantaneous value of current in these conductors. Actually, the two parallel conductors are two halves of a single metallic ribbon, the ends being soldered to terminals and the center looped over a tiny ivory pulley. This pulley is attached to a spring which keeps the ribbons very tight so as to respond to as high a frequency of current as possible. Accurately machined ivory bridges space the ribbon properly in the 0.032-inch air gap. The galvanometer is of great sensitivity, being able to follow as

high frequencies as most of the more expensive oscillographs, has a very light permanent magnet giving a greater field strength than cumbersome electromagnets, and is so small that two such units, complete, may be held in the palm of a man's hand.

The magnet and the vibrator are placed in a cavity in a block of mica with the intervening space filled with a transparent oil which acts as a damping fluid to prevent the vibrator from moving after the current ceases. Thus the vibrator follows rapid changes in electric currents so as to give good results with frequencies as high as 5,000 cycles per second. This cavity is sealed tightly, thus greatly increasing the portability of the outfit.

The light from the straight-line filament of the 2-watt lamp enters the lens window (Fig. 335), strikes the vibrator mirror, is reflected back so as to pass through the lens window again, and is focused on the cylindrical condensing lens a short distance from the photographic film or ground glass, as shown. A reflecting prism between the galvanometer and the cylindrical lens adds to the compactness of the instrument without reducing the length of the optical image. The cylindrical lens condenses a fraction of the length of the straight-line image to a point on the focal plane, thus giving a brilliant spot of light on the ground glass or photographic film. The zero position of this spot of light on the film may be shifted by shifting the lamp slightly in the opposite direction.

The optical system is so efficient that a few watts applied to the lamp gives as intense a wave, for photographing or viewing, as is obtained in arc-lamp oscillographs requiring more than a hundred times that energy from a direct-current circuit. The wave lines are very fine on the film and are individually marked by a cross-hair mechanism, to show which is from circuit 1, which from circuit 2, etc.

The oscillograph has been used to show very minute vibratory movements in stationary and in rotating apparatus. Longitudinal movements, of a shaft, of ten-millionths of an inch, at frequencies as high as 2,000 cycles per second, have been recorded on oscillograms. Also records of torsional vibrations of considerably less than 0.01 degree have been clearly shown without being affected by the high-speed rotation of the same shaft.

CHAPTER XXV

OUTLINES OF SUGGESTED TESTS

1. Calibration and Adjustment of Pressure Gage.—Reference, pages 14 to 27.

Apparatus.—Dead-weight gage tester, standard weights, and gage to be tested.

Method.—Take readings at intervals of 5 pounds per square inch, up and down. Spin platform and weights to eliminate friction. Remove the indicating needle with special jack and take off dial. Sketch parts in interior of gage. Reset needle to read correctly in part of scale most used.¹ Attach needle firmly. Repeat readings up and down for new calibration.

Report.—Explain methods of adjustment. Tabulate as on page 26.

*Curves.*²—See page 26. Draw curves only for final condition of gage.

2. Calibration and Adjustment of Vacuum Gage.—Reference, pages 27 to 28 and 266.

Apparatus.—Mercury U-tube, aspirator (ejector) or air pump, and gage to be tested.

Method.—Take readings at intervals of 2-inch vacuum, up and down. If there is water or other impurity on mercury column, correction must be made. (Reset needle if so instructed.)

Report.—Explain details of method used. Tabulate as on page 26, omitting second column and writing "inches vacuum" for "pressure, pounds per square inch."

Curves.—Similar to instructions for test 1.

3. Thermometer Calibration for Range above 212°F.—Reference, pages 49 to 59.

Apparatus.—Steam gage used in test 1, thermometers (a standard high-reading thermometer is sometimes used to check the results), barometer, and steam tables (Appendix).

Method.—Read thermometer being tested (and "standard" if one is used) at intervals of approximately 5 pounds per square inch on the gage calibrated in test 1. Be sure the steam is not superheated and allow at least 5 minutes after final adjustments of valves before readings are taken.

Report.—Sketch with a simple line drawing the interior of apparatus used with necessary piping connections. Tabulate as on page 59. Calculate "stem" corrections when necessary.

Curves.—(See page 58.) "True" temperatures are from steam tables.

4. Use and Calibration of a Planimeter.—Reference, pages 99 to 112, 152.

Apparatus.—Polar planimeter, large compass, scale, and micrometer.

Method.—1. Measure length of tracing arm from pivot to tracing point and diameter of rolling or graduated wheel to check accuracy for reading areas in square inches.

2. Calculate length of tracing arm so that one revolution of graduated wheel will indicate 8 square inches.

¹ If other instructions are not given assume two-thirds maximum graduation of dial.

² Arrangement of coordinates for curves is throughout to make them most applicable for use; that is, in the use of a curve the given quantity should be read on the scale of abscissas and the value to be found will then be obtained from the scale of ordinates.

3. Find area of zero circle by at least two methods (see pages 102, 104).

4. Determine average error in percentage of instrument by first measuring and then calculating the area of circles of 1-, 2-, and 3-inch diameter (see page 111, footnote). Measure each area three times and take average. Mark percentage error + if instrument reads too small and - if too large.

5. Determine indicated horsepower for an indicator card. Scale of indicator spring = 40 pounds per square inch (that is, No. 40 spring); area of piston = 66 square inches; length of stroke = 1 foot; and r.p.m. = 200.

Report.—Record data, method of measurements, and calculations. Tabulate results.

5. Calibration of Indicator Springs in Compression.—Reference, pages 116 to 123, 129 to 134, 135, and 146. Read Precautions for Care of Indicator (pp. 123 to 125).

Apparatus.—Indicator, set of springs, and indicator spring tester.

Method.—1. Test perpendicularity of motion of pencil point of indicator to atmospheric line.

2. Obtain at least two good calibration cards for each spring similar to Fig. 112, for increasing and decreasing pressures. Obviously, unless the cards obtained for a spring are alike, the work is not successful. Take increments of 5 pounds per square inch for all springs up to and including the "40-pound" spring. For springs of higher scale take increments of 10 pounds per square inch. Lines of maximum pressure on calibration card should be about $1\frac{1}{2}$ inches above the atmospheric line.

When a plunger type of tester is used, that is, similar in principle to Fig. 109, the diameter of the plunger must be measured with a micrometer and the relation accurately calculated between the weight and the unit pressure applied to the indicator in pounds per square inch.

Examine at least two types of indicators. Insert the spring and study the adjustment of the height of the pencil.

When work with an indicator is finished, always remove the "piston" spring and thoroughly clean all parts, inside and outside.

Report.—Tabulate and draw curves as directed on page 135. Calculate the true scale of spring from the average of four equidistant points on this curve. Explain calculations. Discuss any discrepancies in data.

If a "40-pound" spring has been calibrated, assume this was used in obtaining the indicator card used in test 4, and calculate the corrected indicated horsepower.

5a. Calibration of Indicator Springs in Tension (for Use with Vacuum).—Reference, pages 134 to 135.

Apparatus.—Indicator, springs, and special tester.

Method and Report.—Same as for test 5.

6. Study of Reducing Motions.—Reference, pages 136 to 146.

Apparatus.—Steel scale and drawing instruments.

Method and Report.—Examine reducing motions in laboratory. Design an accurate device for an engine as designated by instructor.

7. Steam Calorimeter Tests, Use and Comparison.—Reference, pages 75 to 79.

Apparatus.—Separating calorimeter, glass beaker (or graduate), pail with cover having a hole for insertion of hose for condensing steam, platform scales, watch, throttling calorimeter, thermometers, steam pressure gage, barometer, and monkey wrench.

Method.—Calibrate steam gage. Connect the separating and throttling calorimeters by means of standard sampling nozzles (see page 77) to the same vertical steam pipe where they will both take steam of the same quality. Allow steam to blow through both calorimeters until the temperatures in the throttling type have reached a maximum value and the other has become thoroughly heated, and enough water has collected to bring the level in the water-gage glass up to, or a little above, the zero on the scale. Make the condensing pail about two-thirds full of water. Obtain this

weight of water by weighings. When all conditions are satisfactory, put the hose through which steam has been discharging from the separating calorimeter into the hole in the cover of the pail, and at the same time observe reading of scale of water gage. Read temperatures and pressures every 3 minutes. Run the test until an appreciable volume of steam discharges from the hole in the cover of the pail. Then throw steam tube out of pail and read the level in the water gage. Again weigh pail and contents. Calibrate the scale of the water gage by removing and weighing water from the separating calorimeter between any two levels within the limits of the scale.

Make one test at each of four steam pressures above 60 pounds per square inch gage as designated by instructor.

Calculate the quality of steam roughly by the charts on page 81 or 83 during the progress of the tests and immediately check with data from separating calorimeter.

Report.—Tabulate all observed and calculated data. Sketch and describe fully the calorimeters used. State in detail all operations in performing test. Discuss relative accuracy of results.

8. Test of Platform Scales.—Reference, pages 180 and 181.

Apparatus.—Platform scales to be tested, 12-inch steel scale, graduated to $1\frac{1}{8}$ inch and standard 50- or 100-pound weight.

Method.—Platform scales are probably used more in engineering work about a power plant than any other measuring device, and usually young engineers do not very well understand their operation. They consist essentially of a device by which a load is applied to a system of levers, of long and short arms, arranged so that a load on the platform can be balanced by weights applied at the end of a final lever called the beam, or by shifting a poise along the length of this latter lever. Essentially, it is like the weighing devices shown in Fig. 183. This weighing beam is usually placed on an upright post at one end of the platform.

1. Take off the platform and measure the length of all the lever arms between knife edges to the nearest $\frac{1}{16}$ inch. Draw simple line sketch showing all arms and lengths.

2. Observe the means provided to adjust the beam to read zero.

3. Observe the sensitiveness of the scales by finding the range through which the poise can be shifted without appreciably disturbing the balance.

4. Calibrate the scales after assembling by placing standard weights on the platform and observing the reading on the beam when balanced. After calibration shift scales around roughly and observe result by a recalibration. Test also by placing loads in middle of platform as a scales should be used, and then at any of the sides.

Report.—1. Plot results of the calibration with observed weights as abscissas and standard weights as ordinates.

2. Discuss effect of rough handling on the calibration, and whether an accurate calibration of a scales made before shipping can be considered absolutely reliable later. Explain effect of noncentral loading on the platform.

3. From the measured lever arms calculate the weight of poise for two different positions on the beam.

4. Calculate the weight of a poise for an additional beam which would indicate readings to one-tenth the smallest division on the beam of this scales. (Note that the additional weight of the extra beam would be balanced by a larger adjusting counterweight.)

9. Oil Tests: Viscosity, Flash and Burning Points, and Specific Gravity.—Reference, pages 520 to 530, 539 to 543.

Apparatus.—Viscosimeter, flash tester, hydrometer, two thermometers, two glass graduates, Bunsen burner, matches, wax tapers, test tube, and watch.

Method.—1. Determine flash and burning points of oil in flash tester, with top closed for flash point and open for burning point. To check, use at least two samples

of the same kind of oil. Never use a second time a sample that has been heated. Why? Determine the chill point of the oil if ice is available.

2. With the orifice of viscosimeter and inner cup thoroughly cleaned determine the time required for 60 cubic centimeters of water at "room" temperature to flow through this orifice, starting from the level of the tip of the hook gage. Make three tests and take average. Determine similarly the time required for 60 cubic centimeters of the oil at temperatures of 80, 100, 130, and 160°F. to flow through the same orifice when starting at the same level. Make this test also at 210°F. if steam heating is available.

3. Determine the specific gravity of the oil in both the Baumé and the "specific-gravity" scales at about 80, 100, 120, and 140°F. also at 160 and 210 °F. for steam cylinder and gas engine oils. (Avoid pouring hot oils into cold glass vessels, as they are likely to be broken.)

Report.—Tabulate results of the various tests. Plot (1) temperatures as abscissas and viscosities as ordinates, (2) temperatures as abscissas and specific gravities as ordinates, both curves having the same abscissas on the same sheet.

Discuss suitability or unsuitability of this oil for various services.

10. **Oil Tests (Continued): Coefficient of Friction and Temperature Rise of Bearings.**—Reference, pages 543 to 545.

Apparatus.—Pendulum oil tester, thermometer, wooden strut with knife edge at end, sensitive platform scales, spirit level, and steel scale graduated to $\frac{1}{16}$ inch.

Method.—Support the pendulum in a horizontal position (as determined by a spirit level) on a strut, provided with a knife edge at the bottom where it rests on a platform scales. Determine the weight of the pendulum alone in this position. Measure also the effective length of the lever arm from the center of the journal to a vertical line through the knife edge. Repeat weighing and measurement of length of arm for several points along the pendulum. Remove the pendulum and determine its weight accurately. Measure the length l and diameter d of the journal so as to calculate the projected area $2ld$ of the bearing surface. Compute the constant $\frac{W'R}{\tau}$, and by

setting the pendulum at various angles determine whether the scale is correctly graduated. In order to get satisfactory results the bearing and shaft must be perfectly clean and smooth. Calibrate spring in pendulum.

Operate the machine at constant speed with pressures of 50, 100, 150, and 200 pounds per square inch on the bearing. Record the arc of deflection, temperatures of bearing and of room, speed, and rate of oil feed (usually three drops per minute).

Report.—Plot curves of bearing pressure (pounds per square inch) as abscissas and coefficients of friction and temperatures as ordinates.

Calculate velocity of rubbing faces in feet per minute. In machines arranged like ordinary shop bearings where all the load is on the bottom half of the bearing this velocity is calculated on the basis of only half the circumference.

Discuss possible errors in the data as shown by the curves.

11. **Proximate Analysis of Coal.**—Reference, pages 253 to 257, 258 to 265.

Apparatus.—Crucibles, Bunsen burners, matches, coal crusher, mortar and pestle, 20-mesh sieve, two airtight bottles, drying oven, air-blast lamp (or an equivalent), rubber tubing, watch, desiccator, and "chemical" balance sensitive to 1 part in 1,000 of amount weighed.

Method.—That of American Chemical Society (pages 254 to 256) or of A.S.M.E. (pages 256 and 258) as directed by instructor. Inquire about location of mine from which coal was taken. Make duplicate determinations to check values of moisture and volatile matter.

Report.—Record all data. Determine percentages of moisture, volatile matter, fixed carbon and ash in coal "as received." Also percentages of volatile matter, fixed

carbon, and ash in dry coal. Discuss results by comparing with analyses of coals from same district as given in mechanical engineers' handbooks, etc.

12. Calorific Value of Coal.—Reference, pages 222 to 239.

A. Apparatus.—Bomb calorimeter with platinum crucible, mortar and pestle, 100-mesh sieve, oxygen tank with pipe connections to fit threads on bomb, accurate calorimeter, thermometer, pail,¹ scales for weighing water, fine iron wire for ignition, "chemical" balance sensitive to $\frac{1}{1,000}$ gram, briquetting machine,² monkey wrench, and calorimeter spanner.

B. Apparatus.—Same as *A* except Parr calorimeter is used instead of bomb type, and absolutely pure sodium peroxide is used instead of oxygen gas.

Method.—See pages 222 to 232 for *A* and pages 232 to 239 for *B*. Make at least two determinations.

Report for A or B.—Describe apparatus used and procedure in detail. Calculate heating value of coal tested in B.t.u. per pound "as received," also heating value per pound dry coal and per pound combustible in same units. Record all data. Discuss results by comparing with heating value of coals from same district as given in mechanical engineers' handbooks, etc.

13. Calorific Value of Gas.—Reference, pages 239 to 249.

Apparatus.—Junkers calorimeter (with gas burner), two calorimeter thermometers, two ordinary thermometers, two glass graduates, large pail, platform scales, "wet" gas meter, gas regulator, glass U-tube for gas pressure, barometer, and rubber tubing.

Method.—See pages 239 to 249. Read thermometers every 2 minutes.

Make at least two determinations, which should check within 1 per cent.

Report.—Tabulate all observations. Explain calculations.

Determine "higher" and "lower" heating values of the gas per cubic foot (1) at conditions of test and (2) at standard conditions (see page 241).

Sketch apparatus used. Discuss possible errors in method.

14. Calorific Value of Oil.—Reference, pages 250 to 251.

Apparatus.—Junkers calorimeter (with oil lamp and chemical balance for its attachment) and hydrometer. Otherwise same as for test 13.

Method.—See pages 239 to 251. Collect water from calorimeter during time required for burning exactly 50 grams of oil. Determine specific gravity of oil used. Make at least two determinations.

Report.—Tabulate all observations. Calculate "higher" and "lower" heating values per pound of oil. Discuss result by comparing with data given in books on gas and oil engines or in mechanical engineers' handbooks.

15. Analysis of Flue Gas.—Reference, pages 266 to 275, 279 to 283, 330 to 331.

Apparatus for sampling and chemical absorption of gases as directed by instructor.

Method.—(See pages 268 to 276). Make at least two analyses from the same sample of gas which should check throughout within $\frac{1}{10}$ per cent.

Report.—Tabulate data and results (calculated as percentages).

From the average results of the analysis calculate number of pounds of air required to burn (1) a pound of carbon and (2) a pound of dry coal, assuming the dry coal contains 83 per cent carbon, 3 per cent hydrogen, 4 per cent oxygen, and 10 per cent earthy matter.

16. Study of Brakes.—Reference, pages 158 to 164.

Apparatus.—Steel scale, calipers, and drawing instruments.

Method and Report.—Design Prony brake of type stated by instructor for absorbing brake horsepower at . . . revolutions per minute. Show calculations for diameter

¹ Instead of the pail and scales a large glass graduate is often used to measure the water in cubic centimeters.

² Briquetting the coal is not required by A.S.M.E. recommendations.

at root of thread of tightening bolt, and for determining capacity of scales required to take the pressure of the brake. Calculate also weight of water required per hour for cooling the brake, assuming 10 per cent of heat dissipated by radiation.

Answer the following questions:

1. If water were shut off from the brake what damage would result?
2. Why is it a very bad practice to stop the engine with the full load on the brake?
3. Why should a safety cord be attached to the arm of the brake?
4. What is the effect of putting oil on the rim of the brake pulley? Of putting on water?

If there is any doubt as to the proper answers, run an engine with a Prony brake attached to find out by experience.

17. Calibration of a Transmission Dynamometer.—Reference, pages 172 to 176.

Apparatus.—Dynamometer and its weights, steel scale, hand speed counter, and watch.

Method.—Measure length of lever arm. Observe condition of dashpot as explained on page 175. Remove the brake from its pulley and make a series of runs; that is, without load at various speeds and observe the corresponding readings of instrument. Attach the brake to its pulley and make a series of tests at three different speeds with net loads on the brake for each speed of approximately (1) . . . (2) . . . (3) . . . and (4) . . . pounds. Determine "zero" load or tare of brake, pounds, and length of brake arm, feet.

For each test record: (1) Gross brake load, in pounds, (2) net brake loads, in pounds, (3) reading of dynamometer (theoretically, in pounds), (4) revolutions per minute, and (5) names of observers.

Report.—Examine construction of dynamometer and sketch arrangement of lever arms (and gears, if any). For each speed calculate foot-pounds per minute corresponding to readings of dynamometer and plot curve with these as abscissas and foot-pounds by brake as ordinates. Plot also curve for each speed of reading of dynamometer as abscissas and net brake load, in pounds, as ordinates.

18. Belting Tests.—Reference, pages 517 to 518.

Apparatus.—Belt tester, steel tape, two-hand speed counters, watch, calibrated scales for weighing, and 3-foot scale for lengths.

Method.—(See pages 517 to 518.) Determine "zero" load or tare on each scales. Adjust the distance between driving and driven pulleys when at rest so that there is initial tension in belt of 25 pounds per inch of width. Run tests at a given constant speed of the driver at varying loads until the belt begins to slip. Make about five runs for this value of initial belt tension. Each test should be of at least 5-minute duration during which time both speed counters should be read continuously. Observe for each test also average brake load, pounds, and tension reading, in pounds.

Make similar sets of runs with initial belt tensions of 50 and 75 pounds per inch of width.

Measure diameter of either driving or the driven pulleys, inches, and also arc of contact of belt on this pulley, in inches. Calculate ratio of arc in inches to radius in inches. Measure length, width, and thickness of belt.

Report.—Calculate slip of belt, coefficient of friction, brake horsepower, and efficiency of transmission.

For each value of initial tension plot brake horsepower as abscissas and as ordinates: (1) slip, in per cent, and (2) coefficient of friction. Also initial tension of belt as abscissas and brake horsepower as ordinates.

Discuss results in their application to shop practice.

19. Test of Hoists.—Reference, pages 513 to 517.

Apparatus.—Hoists to be tested, spring balance, scale with graduations marked very plainly.

Method.—Same pages as reference.

Report.—Sketch hoists used. On the same curve sheet plot for each hoist a curve of load lifted (pounds) as abscissas and efficiency as ordinates.

20. Mechanical Efficiency Test of Steam Engine. Determinations of Indicated and Brake Horsepower.—Reference, pages 145 to 152, 158 to 164, 180, 332.

Apparatus.—Steam-engine indicators, steam pressure gage, Prony or rope brake, hand speed counter, watch, scale about 3 feet long, platform scales and planimeter, cans of cylinder and engine oils.

Method.—Put spring in indicator, oil its piston with cylinder oil and the joints of pencil motion with porpoise or similar light oil. Adjust indicator parts so that there is no lost motion in pencil mechanism. Attach firmly to engine cylinder. Adjust cord so that it will have normal tension when engine is turned over both dead centers. Fill engine cylinder lubricator with cylinder oil and all oil cups with engine oil. Adjust feed of all oiling devices. Measure effective brake arm and "zero" load or tare of brake (see pages 159 to 164).

Vary net load on brake by increments of 50 pounds up to maximum engine will carry without slowing down. Run each test for 12 minutes, taking all indicator cards and readings of revolutions per minute and gross brake reading, in pounds, every 3 minutes. Measure as many indicator cards as possible while test is in progress and compare at least a few values of indicated with brake horsepower before test is finished. This is done to check the accuracy of the work. **Always clean the engine and shut off the lubricator and oil cups when finishing a test.**

Calculate engine and brake constants (see pages 152 and 158).

Report.—Tabulate data and calculated results as on page 332. Examine and sketch the reducing motion. Explain whether or not it gives an accurate reduction.

Plot with indicated horsepower as abscissas the following as ordinates: (1) brake horsepower, (2) mechanical efficiency, in per cent, (3) revolutions per minute, and (4) friction horsepower. If friction horsepower has not constant values, discuss reasons.

21. Setting of Plain D-slide Valve on Steam Engine. Reference, pages 333 to 340.

Apparatus.—Steel scale, monkey wrench, trammels or machinist's large dividers, chalk, drawing board and instruments, and indicators.

Method.—(See pages 333 to 338.) Remove steam-chest cover and valve from its stem. Measure face of valve and ports. Make dimensioned drawing like Fig. 257 (page 333) and Zeuner valve diagram. Adjust laps and eccentric as explained in reference. Zeuner diagrams are most useful for valve setting and Bilgram diagrams are best for designing.

Replace steam-chest cover. Attach indicators and take diagrams. Compare with ideal diagram obtained from the Zeuner drawing.

If indicator diagrams are not satisfactory, make the adjustments needed.

Report.—Explain in detail procedure. Discuss each step in making adjustments.

22. Setting of Corliss Valve on Steam Engine.—Reference, pages 340 to 344.

Apparatus.—Monkey wrenches, steel scale, plumb bob and line, and indicators.

Method.—Adjust (1) wrist plate, (2) all reach rods for given laps, (3) rocker. Put engine on dead center and set eccentric for a given lead. Readjust reach rods. Take indicator cards and readjust valves.

23. Volumetric Clearance of Engine.—Reference, pages 344 to 345.

Apparatus.—Pails with cocks near the bottom, rubber tubing, funnels, small platform scales for weighing water in pails, trammels, chalk, monkey wrench, watch, wooden blocks to cover parts of engine, and rubber packing.

Method.—Remove steam-chest cover and cover ports with blocks on top of rubber packing. Set engine on dead center (see page 344). Continue procedure as in reference above.

24. Boiler Test.—Reference, pages 304 to 331.

Apparatus.—Steam gage, draft gage, barometer, watch, wrenches, steam calorimeter (with manometer if needed), thermometers with maximum graduation below 240°F. for (1) external air, (2) boiler room, (3) feed water entering boiler¹ and (4) make-up water; above 240°F. for (1) temperature of steam at steam nozzle (discharge from boiler) and (2) steam calorimeter, platform scales for (1) coal, (2) ashes, and (3) feed water, large tanks for feed water, standard weights, thermocouple for flue temperature, flue-gas apparatus for sampling and analyzing, jars or cans for samples of coal and ash, means for marking level of water in gage glass, large closed cans for accumulating samples of coal and ashes.

Method.—See reference. Calibrate gages and thermometers. Plot a graphical log as test proceeds (see page 293). Run boiler-leakage test for 3 hours at normal boiler pressure before regular boiler test begins.

Report.—Tabulate all data. Use A.S.M.E. "short" or "long" form as directed by instructor.

25. For Economy Tests of Steam Engines, Steam Turbines, Complete Steam Power Plants, Gas Engines, Oil Engines, Gas Producers, as Well as Pump Tests, Injector Tests, Air-compressor Tests, Air-lift Tests, Ventilating-fan Test, Refrigerating-plant Tests, Condenser Tests, Feed-water Heater Tests, and Water-cooling Equipment Tests. See detailed instructions and A.S.M.E. codes, pages 335-367, 380-396, 402-405, 409-421, 421-436, 443-448, 448-460, 464-485, 486-501, 501-506, 507-512, 548-567, 568-569, 572-574.

¹ For plants operating with an economizer a thermometer for higher temperatures would be required.

APPENDIX

The following tables (I, II, and III) of the properties of saturated and superheated steam are abstracted from “Steam Tables” by Prof. J. H. Keenan, 1930 ed., by permission of the publisher, The American Society of Mechanical Engineers.

TABLE I.—SATURATED STEAM: PRESSURE TABLE

		Specific Volume			Total Heat			Entropy				
Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Sat. Liquid V _l	Evap. V _g	Sat. Vapor V _g	Sat. Liquid h _l	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _l	Evap. s _{fg}	Sat. Vapor s _g	Abs. Press. Lb./Sq. In.	Temp. Deg. F.
$\frac{1}{4}$ " Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	$\frac{1}{4}$ " Hg	
$\frac{1}{2}$ " Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	$\frac{1}{2}$ " Hg	
1" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1" Hg	
$1\frac{1}{2}$ " Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	$1\frac{1}{2}$ " Hg	
2" Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	2" Hg	
$2\frac{1}{2}$ " Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	$2\frac{1}{2}$ " Hg	
3" Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3" Hg	
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0	
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0	
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0	
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0	
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0	
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0	
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0	
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0	
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0	
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0	
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0	
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0	
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0	
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0	
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696	
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0	
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0	
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0	
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0	
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0	
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0	
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0	
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0	
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0	
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0	
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0	
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0	
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0	
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0	
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0	
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0	
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0	
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0	
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0	
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0	
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0	
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0	
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0	
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0	
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0	
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0	
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0	
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0	
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0	
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0	
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0	
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0	
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0	
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0	
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0	
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0	
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0	
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0	
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0	
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0	
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	96.0	
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0	

TABLE I.—SATURATED STEAM: PRESSURE TABLE (Continued)

Abs. Press. Lb./Sq. In. p	Temp. Deg. F. t	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In. p
		Sat. Liquid v _f	Evap. v _{fg}	Sat. Vapor v _g	Sat. Liquid h _f	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Evap. s _{fg}	Sat. Vapor s _g	
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE I.—SATURATED STEAM: PRESSURE TABLE (Continued)

Abs. Press. Lb./Sq. In. p	Temp. Deg. F. t	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In. p
		Sat. Liquid v _l	Evap. v _{lg}	Sat. Vapor v _g	Sat. Liquid h _l	Evap. h _{lg}	Sat. Vapor h _g	Sat. Liquid s _l	Evap. s _{lg}	Sat. Vapor s _g	
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	250.0
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	260.0
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	270.0
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	280.0
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	290.0
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	300.0
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	320.0
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	340.0
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	360.0
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	380.0
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400.0
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	420.0
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	440.0
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	460.0
480.0	462.80	0.0197	0.9456	0.9653	444.9	759.0	1203.9	0.6441	0.8228	1.4670	480.0
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500.0
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	520.0
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	540.0
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	560.0
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	580.0
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	600.0
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	620.0
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	640.0
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	660.0
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	680.0
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	700.0
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	720.0
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	740.0
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	760.0
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	780.0
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	800.0
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	820.0
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	840.0
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	860.0
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	880.0
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900.0
920.0	534.56	0.0214	0.4635	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	920.0
940.0	537.13	0.0215	0.4520	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	940.0
960.0	539.66	0.0216	0.4409	0.4625	539.6	651.5	1191.1	0.7410	0.6520	1.3930	960.0
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	980.0
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000.0
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050.0
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100.0
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150.0
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200.0
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250.0
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300.0
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350.0
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400.0
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450.0
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500.0
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600.0
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700.0
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800.0
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	1900.0
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	2000.0
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200.0
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	2400.0
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600.0
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	2800.0
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000.0
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	3200.0
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3226.0

TABLE II.—SATURATED STEAM: TEMPERATURE TABLE

Temp. Fahr.	Abs. Press. Lb./Sq. In.	Specific Volume			Total Heat			Entropy			Temp. Fahr.
		Sat. Liquid V _f	Evap. V _{fg}	Sat. Vapor V _g	Sat. Liquid h _f	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Evap. s _{fg}	Sat. Vapor s _g	
32°	0.0887	0.01602	3301	3301	0.00	1073.4	1073.4	0.0000	2.1834	2.1834	32°
34	0.0961	0.01602	3059	3059	2.01	1072.3	1074.3	0.0041	2.1723	2.1764	34
36	0.1041	0.01602	2835	2835	4.03	1071.2	1075.3	0.0081	2.1614	2.1695	36
38	0.1126	0.01602	2632	2632	6.04	1070.2	1076.2	0.0122	2.1505	2.1627	38
40°	0.1217	0.01602	2445	2445	8.05	1069.1	1077.1	0.0162	2.1397	2.1559	40°
42	0.1315	0.01602	2272	2272	10.05	1068.0	1078.0	0.0202	2.1290	2.1492	42
44	0.1420	0.01602	2112	2112	12.05	1066.9	1078.9	0.0242	2.1184	2.1426	44
46	0.1532	0.01602	1965.3	1965.3	14.06	1065.8	1079.8	0.0282	2.1078	2.1360	46
48	0.1652	0.01602	1829.7	1829.7	16.06	1064.7	1080.8	0.0321	2.0974	2.1295	48
50°	0.1780	0.01602	1704.8	1704.8	18.06	1063.6	1081.7	0.0361	2.0870	2.1231	50°
52	0.1918	0.01602	1588.3	1588.3	20.06	1062.5	1082.6	0.0400	2.0767	2.1167	52
54	0.2063	0.01602	1482.4	1482.4	22.06	1061.4	1083.5	0.0439	2.0665	2.1104	54
56	0.2219	0.01603	1383.5	1383.5	24.05	1060.3	1084.4	0.0478	2.0564	2.1042	56
58	0.2384	0.01603	1292.8	1292.8	26.05	1059.3	1085.3	0.0517	2.0464	2.0980	58
60°	0.2561	0.01603	1208.0	1208.0	28.05	1058.2	1086.2	0.0555	2.0364	2.0919	60°
62	0.2749	0.01603	1129.7	1129.7	30.05	1057.1	1087.1	0.0594	2.0265	2.0859	62
64	0.2949	0.01604	1057.1	1057.1	32.04	1056.0	1088.0	0.0632	2.0167	2.0799	64
66	0.3162	0.01604	989.7	989.6	34.04	1054.9	1089.0	0.0670	2.0069	2.0739	66
68	0.3388	0.01605	927.1	927.1	36.03	1053.8	1089.8	0.0708	1.9973	2.0680	68
70°	0.3628	0.01605	869.0	869.0	38.03	1052.7	1090.8	0.0746	1.9877	2.0622	70°
72	0.3883	0.01606	814.9	815.0	40.02	1051.6	1091.7	0.0783	1.9782	2.0565	72
74	0.4153	0.01606	764.8	764.8	42.02	1050.6	1092.6	0.0820	1.9687	2.0507	74
76	0.4440	0.01606	718.0	718.0	44.01	1049.5	1093.5	0.0858	1.9593	2.0451	76
78	0.4744	0.01607	674.5	674.5	46.00	1048.4	1094.4	0.0895	1.9500	2.0395	78
80°	0.5067	0.01607	633.8	633.8	48.00	1047.3	1095.3	0.0932	1.9407	2.0340	80°
82	0.5409	0.01608	595.9	595.9	50.00	1046.2	1096.2	0.0969	1.9316	2.0285	82
84	0.5772	0.01608	560.5	560.5	52.00	1045.1	1097.1	0.1006	1.9224	2.0230	84
86	0.6153	0.01609	527.6	527.7	54.00	1044.0	1098.0	0.1042	1.9134	2.0176	86
88	0.6555	0.01609	497.1	497.1	56.00	1042.9	1098.9	0.1079	1.9044	2.0123	88
90°	0.6980	0.01610	468.5	468.5	58.00	1041.8	1099.8	0.1115	1.8955	2.0070	90°
92	0.7429	0.01610	441.7	441.8	59.98	1040.7	1100.7	0.1152	1.8866	2.0018	92
94	0.7902	0.01611	416.8	416.8	61.97	1039.6	1101.6	0.1188	1.8778	1.9966	94
96	0.8403	0.01612	393.3	393.3	63.96	1038.5	1102.5	0.1224	1.8691	1.9915	96
98	0.8930	0.01612	371.4	371.4	65.94	1037.4	1103.4	0.1260	1.8604	1.9864	98
100°	0.9487	0.01613	350.8	350.8	67.93	1036.3	1104.2	0.1295	1.8518	1.9813	100°
105	1.1009	0.01615	305.0	305.0	72.91	1033.5	1106.4	0.1384	1.8305	1.9689	105
110	1.274	0.01616	265.8	265.8	77.89	1030.8	1108.6	0.1472	1.8095	1.9567	110
115	1.470	0.01618	232.3	232.3	82.89	1027.9	1110.8	0.1559	1.7889	1.9448	115
120°	1.692	0.01620	203.5	203.6	87.88	1025.1	1113.0	0.1646	1.7685	1.9331	120°
125	1.941	0.01622	178.9	178.9	92.87	1022.2	1115.1	0.1731	1.7485	1.9216	125
130	2.221	0.01625	157.62	157.64	97.86	1019.4	1117.2	0.1816	1.7288	1.9104	130
135	2.536	0.01627	139.15	139.17	102.85	1016.5	1119.3	0.1901	1.7094	1.8995	135
140	2.887	0.01629	123.20	123.22	107.84	1013.6	1121.4	0.1984	1.6903	1.8887	140
145°	3.280	0.01632	109.29	109.31	112.84	1010.6	1123.5	0.2067	1.6715	1.8782	145°
150	3.716	0.01634	97.21	97.23	117.84	1007.7	1125.5	0.2149	1.6530	1.8679	150
155	4.201	0.01637	86.64	86.66	122.85	1004.7	1127.6	0.2231	1.6347	1.8578	155
160	4.739	0.01639	77.38	77.40	127.85	1001.8	1129.6	0.2312	1.6168	1.8479	160
165	5.334	0.01642	69.26	69.28	132.85	998.8	1131.7	0.2392	1.5991	1.8383	165
170°	5.990	0.01645	62.13	62.14	137.85	995.8	1133.7	0.2472	1.5816	1.8288	170°
175	6.716	0.01648	55.81	55.82	142.86	992.8	1135.7	0.2551	1.5644	1.8195	175
180	7.510	0.01650	50.26	50.28	147.87	989.8	1137.7	0.2629	1.5475	1.8105	180
185	8.382	0.01654	45.35	45.36	152.87	986.8	1139.7	0.2707	1.5308	1.8016	185
190	9.336	0.01656	40.99	41.01	157.89	983.8	1141.7	0.2785	1.5144	1.7929	190
195°	10.385	0.01660	37.10	37.12	162.91	980.8	1143.7	0.2862	1.4982	1.7844	195°
200	11.525	0.01663	33.65	33.67	167.94	977.7	1145.6	0.2938	1.4822	1.7760	200
205	12.769	0.01665	30.57	30.59	172.97	974.6	1147.6	0.3014	1.4664	1.7678	205
210	14.123	0.01669	27.82	27.83	177.99	971.5	1149.5	0.3089	1.4508	1.7597	210
215	14.696	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	215
215°	15.591	0.01673	25.35	25.37	183.02	968.3	1151.3	0.3164	1.4353	1.7517	215°
220	17.188	0.01676	23.14	23.16	188.06	965.1	1153.1	0.3238	1.4200	1.7439	220
225	18.915	0.01680	21.15	21.17	193.09	961.8	1154.9	0.3313	1.4049	1.7362	225
230	20.78	0.01683	19.371	19.388	198.15	958.6	1156.7	0.3386	1.3900	1.7286	230
235	22.80	0.01687	17.761	17.778	203.21	955.3	1158.5	0.3460	1.3752	1.7212	235

TABLE II.—SATURATED STEAM: TEMPERATURE TABLE (Continued)

Temp. Fahr. t	Abs. Press. lb./sq. in. p	Specific Volume			Total Heat			Entropy			Temp. Fahr. t
		Sat. Liquid V _l	Evap. V _{ig}	Sat. Vapor V _g	Sat. Liquid h _l	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _l	Evap. s _{ig}	Sat. Vapor s _g	
240°	24.97	0.01690	16.307	16.324	208.26	952.0	1160.2	0.3532	1.3607	1.7138	240°
245	27.31	0.01695	14.991	15.008	213.33	948.6	1161.9	0.3604	1.3462	1.7066	245
250	29.82	0.01698	13.807	13.824	218.39	945.2	1163.6	0.3675	1.3320	1.6995	250
255	32.53	0.01702	12.726	12.743	223.47	941.8	1165.3	0.3746	1.3179	1.6925	255
260	35.43	0.01706	11.745	11.762	228.55	938.4	1166.9	0.3817	1.3040	1.6856	260
265°	38.54	0.01710	10.854	10.871	233.65	934.9	1168.6	0.3888	1.2901	1.6789	265°
270	41.85	0.01714	10.044	10.061	238.74	931.4	1170.1	0.3958	1.2765	1.6723	270
275	45.40	0.01719	9.304	9.321	243.84	927.8	1171.6	0.4028	1.2630	1.6658	275
280	49.20	0.01723	8.626	8.644	248.95	924.2	1173.2	0.4097	1.2496	1.6593	280
285	53.25	0.01728	8.007	8.024	254.08	920.6	1174.7	0.4166	1.2363	1.6529	285
290°	57.55	0.01732	7.442	7.459	259.20	917.0	1176.2	0.4234	1.2233	1.6467	290°
295	62.13	0.01737	6.923	6.940	264.34	913.4	1177.7	0.4303	1.2103	1.6406	295
300	67.01	0.01742	6.446	6.464	269.48	909.6	1179.1	0.4370	1.1974	1.6345	300
305	72.18	0.01747	6.009	6.026	274.64	905.8	1180.4	0.4438	1.1847	1.6285	305
310	77.68	0.01752	5.606	5.623	279.80	902.1	1181.9	0.4505	1.1721	1.6226	310
315	83.50	0.01757	5.234	5.252	284.98	898.2	1183.2	0.4571	1.1596	1.6167	315
320°	89.65	0.01762	4.892	4.910	290.17	894.4	1184.6	0.4638	1.1472	1.6110	320°
325	96.16	0.01768	4.573	4.591	295.38	890.5	1185.9	0.4705	1.1349	1.6054	325
330	103.03	0.01773	4.285	4.303	300.59	886.5	1187.1	0.4770	1.1227	1.5998	330
335	110.31	0.01779	4.016	4.034	305.81	882.5	1188.3	0.4836	1.1106	1.5942	335
340	117.99	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	340
345°	126.10	0.01791	3.535	3.553	316.30	874.3	1190.6	0.4966	1.0866	1.5832	345°
350	134.62	0.01797	3.321	3.338	321.55	870.2	1191.8	0.5031	1.0748	1.5779	350
355	143.58	0.01804	3.121	3.139	326.82	866.0	1192.8	0.5095	1.0630	1.5725	355
360	153.01	0.01810	2.936	2.954	332.10	861.7	1193.8	0.5160	1.0514	1.5673	360
365	162.93	0.01817	2.764	2.782	337.40	857.4	1194.8	0.5224	1.0397	1.5621	365
370°	173.33	0.01823	2.604	2.622	342.71	853.0	1195.7	0.5288	1.0282	1.5570	370°
375	184.23	0.01830	2.455	2.473	348.05	848.6	1196.7	0.5352	1.0167	1.5519	375
380	195.70	0.01836	2.315	2.333	353.39	844.1	1197.5	0.5415	1.0053	1.5468	380
385	207.71	0.01844	2.185	2.203	358.76	839.5	1198.3	0.5479	0.9939	1.5418	385
390	220.29	0.01850	2.0631	2.0816	364.14	834.9	1199.0	0.5542	0.9826	1.5368	390
395	233.47	0.01858	1.9489	1.9675	369.54	830.3	1199.8	0.5605	0.9714	1.5319	395
400°	247.25	0.01865	1.8421	1.8608	374.96	825.5	1200.4	0.5668	0.9602	1.5270	400°
410	276.72	0.01880	1.6493	1.6681	385.86	815.8	1201.6	0.5792	0.9381	1.5173	410
420	308.82	0.01896	1.4792	1.4982	396.84	805.8	1202.7	0.5916	0.9161	1.5077	420
430	343.71	0.01911	1.3295	1.3486	407.91	795.5	1203.4	0.6039	0.8942	1.4982	430
440	381.59	0.01928	1.1965	1.2158	419.07	784.9	1203.9	0.6162	0.8724	1.4887	440
450°	422.61	0.0195	1.0782	1.0977	430.3	773.8	1204.1	0.6284	0.8507	1.4792	450°
460	466.94	0.0196	0.9730	0.9927	441.7	762.3	1204.0	0.6407	0.8290	1.4696	460
470	514.76	0.0198	0.8793	0.8991	453.2	750.3	1203.5	0.6530	0.8071	1.4601	470
480	566.26	0.0200	0.7951	0.8151	465.0	737.8	1202.8	0.6654	0.7852	1.4506	480
490	621.67	0.0202	0.7195	0.7398	477.0	724.7	1201.7	0.6779	0.7632	1.4410	490
500°	681.09	0.0205	0.6516	0.6721	489.1	711.1	1200.2	0.6904	0.7410	1.4314	500°
510	744.74	0.0207	0.5903	0.6110	501.6	696.9	1198.4	0.7031	0.7187	1.4218	510
520	812.72	0.0210	0.5347	0.5557	514.2	682.1	1196.3	0.7158	0.6963	1.4121	520
530	885.31	0.0213	0.4845	0.5058	527.0	666.8	1193.8	0.7286	0.6738	1.4024	530
540	962.73	0.0216	0.4394	0.4610	540.0	651.0	1191.0	0.7414	0.6512	1.3926	540
550°	1045.4	0.0219	0.3982	0.4201	553.2	634.5	1187.8	0.7543	0.6285	1.3828	550°
560	1133.4	0.0223	0.3605	0.3828	566.7	617.5	1184.2	0.7672	0.6056	1.3728	560
570	1227.6	0.0227	0.3261	0.3488	580.4	599.7	1180.2	0.7802	0.5825	1.3626	570
580	1327.2	0.0231	0.2949	0.3180	594.4	581.3	1175.7	0.7932	0.5592	1.3524	580
590	1432.7	0.0236	0.2664	0.2900	608.7	562.2	1170.8	0.8064	0.5356	1.3420	590
600°	1544.6	0.0241	0.2401	0.2642	623.2	542.3	1165.5	0.8198	0.5118	1.3316	600°
610	1663.2	0.0247	0.2159	0.2406	638.0	521.4	1159.5	0.8332	0.4875	1.3208	610
620	1788.8	0.0254	0.1933	0.2186	653.4	499.2	1152.5	0.8470	0.4623	1.3093	620
630	1921.9	0.0261	0.1721	0.1982	669.5	474.8	1144.3	0.8612	0.4358	1.2970	630
640	2062.8	0.0269	0.1522	0.1791	686.6	447.9	1134.5	0.8763	0.4073	1.2836	640
650°	2211.4	0.0278	0.1331	0.1610	705.2	417.7	1122.8	0.8924	0.3764	1.2688	650°
660	2368.6	0.0290	0.1148	0.1437	725.3	383.6	1108.8	0.9097	0.3426	1.2523	660
670	2534.2	0.0304	0.0966	0.1269	747.5	344.4	1091.9	0.9287	0.3049	1.2336	670
680	2709.7	0.0322	0.0781	0.1102	772.6	298.5	1071.2	0.9499	0.2619	1.2119	680
690	2896.8	0.0347	0.0589	0.0936	803.0	241.2	1044.2	0.9755	0.2098	1.1852	690
700°	3096.4	0.0394	0.0353	0.0747	846.3	157.0	1003.2	1.0117	0.1354	1.1471	700°
706.1	3226.0	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	706.1

TABLE III.—SUPERHEATED STEAM

Abs. Press. Lb./Sq. In. (Sat. Temp.)			Temperature—Degrees Fahrenheit															
			Sat. Water	Sat. Steam	180°	200°	280°	300°	380°	400°	480°	500°	580°	600°	700°	800°	900°	1000°
1 (101.76)	v	0.02	333.9	362.7	392.5	422.3	452.1	481.9	511.7	541.6	571.3	601.1	630.9	660.6	690.6	750.2	809.8	869.4
	h	69.7	1105.0	1127.0	1149.8	1172.4	1195.0	1217.8	1240.7	1263.9	1287.2	1310.9	1334.6	1358.3	1432.6	1483.3	1535.2	
	s	0.1326	1.9769	2.0144	2.0503	2.0835	2.1142	2.1433	2.1707	2.1968	2.2218	2.2458	2.2688	2.2912	2.3125	2.3335	2.3522	2.4291
6 (162.25)	v	0.02	73.61	78.17	84.19	90.21	96.21	102.19	108.19	114.16	120.14	126.11	132.08	138.05	149.99	161.91	173.83	
	h	130.1	1130.6	1148.2	1171.3	1194.2	1217.2	1240.2	1263.5	1286.9	1310.7	1334.4	1358.9	1432.5	1482.2	1535.1		
	s	0.2348	1.8435	1.8712	1.9050	1.9361	1.9654	1.9930	2.0192	2.0443	2.0683	2.0914	2.1135	2.1351	2.1761	2.2149	2.2517	
10 (193.21)	v	0.02	38.45	38.88	41.92	44.98	48.00	51.01	54.02	57.02	60.02	63.01	68.99	74.96	80.92	86.89		
	h	161.1	1143.0	1146.3	1169.9	1193.2	1216.4	1239.6	1262.9	1286.4	1310.1	1334.0	1358.2	1432.3	1483.1	1535.0		
	s	0.2834	1.7874	1.7925	1.8271	1.8587	1.8838	1.9161	1.9430	1.9676	1.9916	2.0148	2.0586	2.0997	2.1385	2.1753		
14.696 (212.00)	v	0.02	26.82	28.40	30.52	32.59	34.65	36.70	38.75	40.79	42.83	46.91	50.97	55.03	59.09			
	h	180.0	1150.2	1168.6	1192.2	1215.6	1239.0	1262.4	1286.0	1309.7	1333.7	1358.4	1432.1	1482.9	1534.9			
	s	0.3119	1.7564	1.7832	1.8154	1.8452	1.8731	1.8996	1.9249	1.9490	1.9722	2.0161	2.0572	2.0961	2.1330			
20 (227.96)	v	0.02	20.10	20.79	22.36	23.90	25.43	26.94	28.45	29.95	31.46	34.46	37.44	40.43	43.42			
	h	196.1	1156.0	1167.0	1191.1	1214.8	1238.3	1261.9	1285.5	1309.3	1333.4	1382.1	1432.0	1482.8	1534.8			
	s	0.3356	1.7317	1.7474	1.7802	1.8104	1.8386	1.8652	1.8906	1.9148	1.9380	1.9819	2.0232	2.0620	2.0989			
40 (267.24)	v	0.017	10.497	11.044	11.843	12.623	13.392	14.161	14.922	15.682	17.190	18.686	20.18	21.68				
	h	235.9	1169.2	1186.5	1211.3	1235.6	1259.7	1283.7	1307.8	1332.0	1381.2	1431.3	1482.3	1534.4				
	s	0.3919	1.6759	1.6990	1.7309	1.7599	1.7871	1.8128	1.8373	1.8607	1.9050	1.9464	1.9854	2.0224				
60 (292.71)	v	0.017	7.172	7.260	7.821	8.353	8.878	9.398	9.911	10.423	11.435	12.436	13.439	14.440				
	h	262.0	1177.0	1181.2	1207.7	1232.8	1257.4	1281.9	1306.2	1330.7	1379.2	1430.6	1481.8	1533.9				
	s	0.4271	1.6434	1.6488	1.6827	1.7128	1.7406	1.7667	1.7915	1.8151	1.8597	1.9014	1.9406	1.9775				
80 (312.03)	v	0.018	5.470	5.806	6.217	6.618	7.015	7.406	7.793	8.158	8.558	9.313	10.067	10.817				
	h	281.9	1182.4	1203.8	1229.9	1255.2	1280.0	1304.7	1329.3	1353.9	1378.2	1429.9	1481.3	1533.5				
	s	0.4532	1.6202	1.6473	1.6785	1.7070	1.7336	1.7586	1.7824	1.8054	1.8274	1.8694	1.9086	1.9456				
100 (327.83)	v	0.018	4.426	4.594	4.934	5.263	5.585	5.903	6.215	6.531	6.831	7.439	8.044	8.644				
	h	298.3	1186.6	1199.7	1226.9	1252.8	1278.0	1303.0	1327.9	1352.8	1378.2	1429.2	1480.7	1533.1				
	s	0.4742	1.6022	1.6185	1.6512	1.6805	1.7075	1.7329	1.7569	1.7803	1.8023	1.8445	1.8838	1.9209				
120 (341.26)	v	0.018	3.725	3.781	4.077	4.359	4.632	4.899	5.162	5.428	5.680	6.189	6.693	7.196				
	h	312.4	1189.8	1195.2	1223.8	1250.4	1276.1	1301.4	1326.5	1351.7	1377.2	1428.5	1480.2	1532.7				
	s	0.4918	1.5874	1.5940	1.6283	1.6584	1.6859	1.7115	1.7359	1.7591	1.7816	1.8240	1.8635	1.9007				
140 (353.03)	v	0.018	3.216	3.465	3.713	3.951	4.182	4.410	4.637	4.857	5.297	5.728	6.161					
	h	324.7	1192.4	1220.5	1247.9	1274.1	1299.7	1325.1	1350.4	1375.7	1401.0	1452.8	1504.7	1556.6				
	s	0.5070	1.5747	1.6084	1.6393	1.6674	1.6933	1.7179	1.7419	1.7640	1.8066	1.8462	1.8836					
160 (363.55)	v	0.018	2.830	3.005	3.227	3.440	3.645	3.846	4.040	4.227	4.627	5.006	5.385					
	h	335.9	1194.5	1217.1	1245.4	1272.1	1298.0	1323.7	1349.3	1375.2	1401.1	1453.1	1505.1					
	s	0.5205	1.5636	1.5906	1.6224	1.6510	1.6774	1.7022	1.7257	1.7487	1.7915	1.8313	1.8687					
180 (373.08)	v	0.018	2.529	2.646	2.849	3.041	3.226	3.407	3.582	3.760	4.105	4.444	4.782					
	h	346.0	1196.3	1215.5	1242.7	1270.0	1296.3	1322.2	1348.2	1374.2	1426.3	1478.5	1531.4					
	s	0.5327	1.5538	1.5743	1.6073	1.6364	1.6631	1.6882	1.7131	1.7351	1.7782	1.8180	1.8555					
200 (381.82)	v	0.018	2.285	2.358	2.547	2.722	2.892	3.056	3.216	3.376	3.688	3.995	4.299					
	h	355.3	1197.8	1209.8	1240.0	1267.9	1294.6	1320.8	1346.6	1372.1	1425.6	1478.0	1531.0					
	s	0.5438	1.5450	1.5592	1.5934	1.6231	1.6502	1.6756	1.7000	1.7228	1.7662	1.8062	1.8438					
220 (389.89)	v	0.019	2.084	2.122	2.299	2.462	2.617	2.769	2.916	3.062	3.347	3.628	3.903					
	h	364.0	1199.0	1205.9	1237.3	1265.7	1292.9	1319.3	1345.7	1372.1	1424.8	1477.4	1530.6					
	s	0.5540	1.5369	1.5450	1.5805	1.6109	1.6384	1.6641	1.6891	1.7117	1.7553	1.7954	1.8331					
240 (397.40)	v	0.0186	1.9156	1.9250	2.092	2.244	2.389	2.529	2.669	2.800	3.063	3.321	3.574					
	h	372.1	1200.1	1201.9	1234.4	1263.5	1291.1	1317.8	1344.5	1371.0	1424.1	1476.8	1530.2					
	s	0.5635	1.5295	1.5317	1.5684	1.5996	1.6275	1.6534	1.6784	1.7014	1.7453	1.7856	1.8234					
260 (404.43)	v	0.0187	1.7723	1.9165	2.060	2.195	2.327	2.459	2.591	2.719	2.823	3.062	3.295					
	h	379.8	1201.0	1221.5	1251.3	1289.3	1316.3	1343.3	1370.0	1423.3	1476.3	1529.7						
	s	0.5723	1.5227	1.5571	1.5890	1.6174	1.6435	1.6685	1.6919	1.7360	1.7765	1.8144						
280 (411.06)	v	0.0188	1.6490	1.7655	1.9019	2.030	2.153	2.276	2.399	2.517	2.617	2.839	3.056					
	h	387.0	1201.8	1228.4	1259.0	1287.5	1314.8	1342.1	1368.9	1422.5	1475.7	1529.3						
	s	0.5805	1.5163	1.5462	1.5790	1.6078	1.6343	1.6588	1.6830	1.7274	1.7680	1.8060						
300 (417.33)	v	0.0189	1.5414	1.6347	1.7648	1.8863	2.002	2.117	2.224	2.324	2.438	2.646	2.849					
	h	393.9	1202.4	1232.5	1256.7	1285.6	1313.3	1341.0	1367.8	1421.7	1475.1	1528.9						
	s	0.5883	1.5102	1.5359	1.5695	1.5988	1.6256	1.6500	1.6747	1.7193	1.7601	1.7983						
350 (431.71)	v	0.0191	1.3245	1.3712	1.4899	1.5981	1.7003	1.7945	1.8945	2.0899	2.260	2.435						
	h	409.8	1203.6	1217.1	1250.7	1280.9	1309.4	1337.9	1365.1	1419.8	1473.6	1527.9						
	s	0.6061	1.4966	1.5117	1.5477	1.5783	1.6059	1.6311	1.6561	1.7012	1.7424	1.7809						
400 (444.58)	v	0.0194	1.1601	1.1726	1.2828	1.3817	1.4740	1.5600	1.6472	1.8119	1.9704	2.125						
	h	424.2	1204.1	1208.3	1244.3	1276.0	1305.5	1335.0	1362.3	1417.7	1472.1	1526.8						
	s	0.6218	1.4843	1.4892	1.5276	1.5599	1.5884	1.6136	1.6396	1.6854	1.7270	1.7658						

* In this table v is specific volume, cubic feet per pound, h is the heat content (Enthalpy), B.t.u. per pound, and s is Entropy, B.t.u. per degree Fahrenheit per pound.

TABLE III.—SUPERHEATED STEAM (Continued)

Abs. Press. Lb./Sq. In. (Sat. Temp.)		Sat. Water	Sat. Steam	Temperature—Degrees Fahrenheit															
				500°	550°	600°	630°	640°	660°	680°	700°	750°	800°	850°	900°	950°	1000°		
450 (456.27)	v	0.0196	1.0303	1.1204	1.2127	1.2972	1.3299	1.3616	1.3932	1.4242	1.4548	1.5299	1.6032	1.6752	1.7455	1.8148	1.8834		
	h	437.4	1204.1	1237.6	1271.0	1301.5	1313.2	1324.9	1336.5	1348.0	1359.4	1387.8	1415.7	1443.2	1470.6	1498.2	1525.6		
	s	0.6361	1.4732	1.5091	1.5430	1.5725	1.5835	1.5942	1.6046	1.6148	1.6248	1.6487	1.6714	1.6928	1.7133	1.7331	1.7524		
500 (466.99)	v	0.0198	0.9261	0.9905	1.0775	1.1558	1.1861	1.2153	1.2444	1.2727	1.3009	1.3697	1.4365	1.5018	1.5655	1.6284	1.6903		
	h	449.7	1203.7	1230.5	1265.8	1297.3	1309.4	1321.4	1333.2	1344.9	1356.6	1385.3	1413.6	1441.5	1469.1	1496.9	1524.8		
	s	0.6493	1.4630	1.4915	1.5274	1.5579	1.5692	1.5802	1.5908	1.6012	1.6113	1.6356	1.6586	1.6802	1.7009	1.7210	1.7404		
550 (476.92)	v	0.0200	0.8402	0.8823	0.9658	1.0398	1.0679	1.0953	1.1223	1.1487	1.1746	1.2381	1.2996	1.3596	1.4181	1.4756	1.5321		
	h	461.3	1203.0	1222.8	1260.3	1293.1	1305.5	1317.7	1329.8	1341.8	1353.6	1382.8	1411.5	1439.7	1467.6	1495.8	1523.8		
	s	0.6616	1.4536	1.4747	1.5126	1.5443	1.5559	1.5671	1.5780	1.5886	1.5990	1.6236	1.6468	1.6687	1.6897	1.7100	1.7296		
600 (486.17)	v	0.0202	0.7677	0.7922	0.8728	0.9431	0.9695	0.9954	1.0206	1.0452	1.0694	1.1285	1.1855	1.2411	1.2953	1.3483	1.4003		
	h	472.3	1202.1	1214.7	1254.6	1288.7	1301.5	1314.1	1326.4	1338.6	1350.6	1380.3	1409.3	1437.8	1466.1	1494.4	1522.8		
	s	0.6731	1.4447	1.4582	1.4986	1.5316	1.5436	1.5551	1.5662	1.5770	1.5874	1.6125	1.6360	1.6582	1.6794	1.6999	1.7196		
700 (503.04)	v	0.0206	0.6527	0.7251	0.7905	0.8143	0.8376	0.8602	0.8822	0.9038	0.9559	1.0063	1.0554	1.1029	1.1484	1.1929			
	h	492.9	1199.7	1242.4	1279.7	1293.3	1306.5	1319.4	1332.0	1344.5	1357.1	1404.9	1434.1	1463.0	1491.9	1520.8			
	s	0.6943	1.4285	1.4720	1.5080	1.5208	1.5329	1.5444	1.5557	1.5665	1.5923	1.6165	1.6392	1.6608	1.6817	1.7018			
800 (518.18)	v	0.0209	0.5653	0.6128	0.6750	0.6974	0.7189	0.7395	0.7596	0.7791	0.8265	0.8723	0.9157	0.9577	0.9982	1.0374			
	h	511.8	1196.7	1226.8	1270.1	1284.6	1298.6	1312.1	1325.3	1338.2	1369.7	1400.4	1430.3	1459.9	1489.3	1518.8			
	s	0.7135	1.4139	1.4462	1.4862	1.4998	1.5127	1.5248	1.5365	1.5477	1.5744	1.5992	1.6225	1.6446	1.6659	1.6864			
900 (531.95)	v	0.0213	0.4969	0.5234	0.5844	0.6057	0.6258	0.6452	0.6639	0.6821	0.7257	0.7675	0.8072	0.8451	0.8815	0.9166			
	h	529.5	1193.3	1213.6	1259.8	1275.5	1290.3	1304.5	1318.3	1331.7	1364.2	1395.8	1426.5	1456.8	1486.8	1516.8			
	s	0.7311	1.4005	1.4208	1.4656	1.4803	1.4938	1.5067	1.5188	1.5304	1.5579	1.5835	1.6074	1.6301	1.6518	1.6727			
1000 (544.58)	v	0.0217	0.4419	0.4495	0.5111	0.5317	0.5509	0.5692	0.5870	0.6040	0.6449	0.6837	0.7202	0.7547	0.7877	0.8199			
	h	546.0	1189.6	1196.5	1248.7	1265.8	1281.6	1296.6	1311.0	1324.9	1358.5	1391.0	1422.6	1453.6	1484.3	1514.8			
	s	0.7473	1.3881	1.3949	1.4455	1.4615	1.4760	1.4895	1.5022	1.5144	1.5427	1.5691	1.5936	1.6169	1.6390	1.6603			
1100 (556.28)	v	0.0222	0.3960	0.4500	0.4705	0.4893	0.5071	0.5239	0.5401	0.5788	0.6152	0.6490	0.6810	0.7115	0.7408				
	h	561.7	1185.6	1236.6	1255.4	1272.5	1288.3	1303.4	1317.9	1332.6	1386.1	1418.7	1450.4	1481.8	1512.8				
	s	0.7624	1.3765	1.4257	1.4433	1.4590	1.4733	1.4866	1.4993	1.5285	1.5557	1.5810	1.6048	1.6274	1.6491				
1200 (567.14)	v	0.0226	0.3582	0.3985	0.4189	0.4373	0.4547	0.4710	0.4865	0.5233	0.5578	0.5897	0.6195	0.6478	0.6750				
	h	576.5	1181.4	1223.4	1244.2	1262.7	1279.6	1295.5	1310.6	1346.5	1381.1	1414.7	1447.2	1479.3	1510.8				
	s	0.7765	1.3656	1.4058	1.4254	1.4423	1.4576	1.4716	1.4848	1.5151	1.5431	1.5692	1.5937	1.6168	1.6388				
1400 (586.96)	v	0.0235	0.2983	0.3144	0.3361	0.3548	0.3717	0.3874	0.4021	0.4365	0.4678	0.4964	0.5229	0.5476	0.5712				
	h	604.3	1172.4	1192.4	1218.9	1241.3	1260.8	1278.6	1295.2	1333.8	1370.8	1406.5	1440.8	1474.3	1506.9				
	s	0.8024	1.3452	1.3643	1.3891	1.4096	1.4272	1.4429	1.4574	1.4899	1.5199	1.5477	1.5736	1.5976	1.6204				
1600 (604.74)	v	0.0244	0.2528	0.2706	0.2908	0.3083	0.3239	0.3384	0.3708	0.4002	0.4264	0.4503	0.4725	0.4935					
	h	630.2	1162.7	1188.2	1216.2	1239.6	1260.1	1278.6	1320.3	1360.0	1398.1	1434.3	1469.3	1502.9					
	s	0.8262	1.3265	1.3503	1.3760	1.3971	1.4152	1.4313	1.4666	1.4986	1.5283	1.5555	1.5808	1.6042					
1800 (620.86)	v	0.0254	0.2167	0.2384	0.2574	0.2736	0.2880	0.3195	0.3472	0.3716	0.3938	0.4140	0.4330						
	h	654.7	1151.8	1186.3	1215.3	1239.4	1260.6	1306.2	1348.7	1389.5	1427.8	1464.3	1498.7						
	s	0.8482	1.3083	1.3399	1.3661	1.3874	1.4058	1.4444	1.4787	1.5105	1.5392	1.5656	1.5897						
2000 (635.61)	v	0.0265	0.1875	0.1931	0.2145	0.2320	0.2468	0.2781	0.3047	0.3279	0.3486	0.3673	0.3847						
	h	679.0	1139.0	1149.0	1186.6	1216.0	1240.7	1291.4	1337.0	1380.5	1421.1	1459.2	1494.7						
	s	0.8696	1.2896	1.2988	1.3326	1.3586	1.3802	1.4230	1.4599	1.4938	1.5242	1.5518	1.5765						
2200 (649.3)	v	0.0277	0.1623	0.1760	0.1960	0.2120	0.2438	0.2696	0.2919	0.3115	0.3289	0.3451							
	h	703.7	1123.8	1150.9	1188.7	1218.4	1275.7	1324.8	1371.3	1414.2	1454.0	1490.4							
	s	0.8912	1.2700	1.2943	1.3278	1.3536	1.4021	1.4418	1.4779	1.5102	1.5390	1.5644							
2400 (661.9)	v	0.0292	0.1404	0.1637	0.1816	0.2148	0.2401	0.2618	0.2805	0.2969	0.3121								
	h	729.4	1105.8	1155.5	1192.8	1259.1	1312.3	1361.6	1407.1	1448.7	1486.1								
	s	0.9133	1.2488	1.2928	1.3253	1.3813	1.4244	1.4628	1.4969	1.5270	1.5531								
2600 (673.8)	v	0.0310	0.1205	0.1314	0.1540	0.1896	0.2151	0.2361	0.2542	0.2699	0.2842								
	h	756.7	1084.5	1110.0	1162.5	1241.0	1299.2	1351.9	1399.8	1443.2	1481.7								
	s	0.9364	1.2257	1.2481	1.2938	1.3602	1.4074	1.4484	1.4842	1.5157	1.5426								
2800 (684.9)	v	0.0333	0.1021	0.1275	0.1674	0.1934	0.2141	0.2315	0.2466	0.2603									
	h	786.7	1058.9	1123.8	1221.3	1285.5	1341.6	1392.1	1437.6	1477.3									
	s	0.9618	1.1996	1.2559	1.3384	1.3905	1.4342	1.4720	1.5049	1.5326									
3000 (695.3)	v	0.0367	0.0844	0.0983	0.1476	0.1742	0.1947	0.2118	0.2265	0.2396									
	h	823.1	1025.6	1066.3	1199.3	1271.1	1331.0	1384.3	1431.7	1472.9									
	s	0.9922	1.1676	1.2028	1.3155	1.3737	1.4203	1.4602	1.4946	1.5233									
3200 (704.9)	v	0.0459	0.0601	0.1294	0.1572	0.1777	0.1945	0.2088	0.2214										
	h	887.0	962.9	1174.5	1255.9	1320.1	1376.1	1425.6	1468.4										
	s	1.0461	1.1112	1.2907	1.3567	1.4067	1.4487	1.4845	1.5144										
3226 (706.1)	v	0.0522	0.0522	0.1271	0.1552	0.1757	0.1924	0.2067	0.2192										
	h	925.0	925.0	1171.2	1253.8	1318.7	1375.0	1424.9	1467.8										
	s	1.0785	1.0785	1.2874	1.3545	1.4050	1.4472	1.4832	1.5133										
3500	v			0.1042	0.1349	0.15													

TABLE IV.—PROPERTIES OF COMMON SUBSTANCES

Substance	Specific grav-ity	Density		Specific heat	Coefficient of expansion per degree Fahrenheit	
		Wt. per cu. ft., lb.	Wt. per cu. in., lb.		Linear	Volumetric
Air.....				0.238		
Aluminum.....	2.60	161	0.095	0.212	0.000011	0.000033
Bismuth.....	9.82	613	0.353	0.031	0.000008	0.000024
Boxwood, along or across grain.....					0.000002	0.000006
Brass.....	8.10	503	0.293	0.094	0.000001	0.000003
Cement.....	2.24	140	0.083	0.20	0.000008	0.000024
Copper.....	8.79	545	0.318	0.097	0.000009	0.000028
Coal (anthracite).....	1.43	88.7	0.058	0.241		
Coke.....	1.00	62.4	0.037	0.203		
Flue Gas.....				0.24		
Gasoline.....	0.68	42.4				
Glass.....	2.89	180.7	0.105	0.198	0.000005	0.000014
Gold.....	19.26	1,200	0.697	0.032	0.000008	0.000024
Ice (at 32°F.).....	0.92	57.5	0.033	0.504		
Iron (cast).....	7.5	465	0.271	0.130	0.000006	0.000018
Iron (wrought).....	7.74	582	0.280	0.110	0.000007	0.000021
Lead.....	11.35	708	0.411	0.031	0.000016	0.000048
Limestone.....	3.16	197	0.114	0.217		
Mercury (at 32°F.).....	13.60	849	0.492	0.033	0.000033	0.000100
Nickel.....	8.90	547	0.321	0.109	0.000007	0.000020
Pine (white), along grain..	0.55	34	0.020	0.65	0.0000025	0.000008
Pine (white), across grain..					0.000020	0.000080
Platinum.....	21.5	1,342	0.779	0.032	0.000005	0.000015
Porcelain.....					0.000002	0.000006
Oak, along grain.....					0.000003	0.000009
Silver.....	10.47	653	0.379	0.056	0.000011	0.000033
Steel.....	7.83	486	0.292	0.116	0.000007	0.000020
Tin.....	7.29	452	0.264	0.056	0.000012	0.000035
Zinc.....	7.19	445	0.260	0.095	0.000016	0.000048

TABLE V.—METRIC CONVERSION TABLE

Millimeters $\times$ 0.03937 = inches.
Millimeters $\div$ 25.4 = inches.
Centimeters $\times$ 0.3937 = inches.
Centimeters $\div$ 2.54 = inches.
Meters $\times$ 39.37 = inches. (Act of Congress.)
Meters $\times$ 3.281 = feet.
Meters $\times$ 1.094 = yards.
Kilometers $\times$ 0.6214 = miles.
Kilometers $\div$ 1.6093 = miles.
Kilometers $\times$ 3,280.8 = feet.
Square millimeters $\times$ 0.00155 = square inches.
Square millimeters $\div$ 645.2 = square inches.
Square centimeters $\times$ 0.155 = square inches.
Square centimeters $\div$ 6.452 = square inches.
Square meters $\times$ 10.764 = square feet.
Square kilometers $\times$ 247.1 = acres.
Hectare $\times$ 2.471 = acres.
Cubic centimeters $\div$ 16.387 = cubic inches.
Cubic meters $\times$ 35.314 = cubic feet.
Cubic meters $\times$ 1.308 = cubic yards.
Cubic meters $\times$ 264.2 = gallons (231 cubic inches).
Liters $\times$ 61.023 = cubic inches. (Act of Congress.)
Liters $\times$ 0.2642 = gallons (231 cubic inches).
Liters $\div$ 3.785 = gallons (231 cubic inches).
Liters $\div$ 28.317 = cubic feet.
Hectoliters $\times$ 3.531 = cubic feet.
Hectoliters $\times$ 2.838 = bushels (2,150.42 cubic inches).
Hectoliters $\times$ 0.1308 = cubic yards.
Hectoliters $\times$ 26.42 = gallons (231 cubic inches).
Grams $\times$ 15.432 = grains. (Act of Congress.)
Grams $\times$ 981 = dynes.
Grams $\div$ 28.35 = ounces avoirdupois.
Grains $\div$ 15.432 = grams.
Grains $\div$ 7,000 = pounds.
Joule $\times$ 0.7373 = foot-pounds.
Kilograms $\times$ 2.2046 = pounds.
Kilograms $\times$ 35.27 = ounces avoirdupois.
Kilograms $\div$ 907.2 = tons (2,000 pounds).
Kilograms per square centimeter $\times$ 14.223 = pounds per square inch.
Kilogram-meters $\times$ 7.233 = foot-pounds.
Kilo per square meter $\times$ 0.672 = pounds per square foot.
Kilo per cubic meter $\times$ 0.0624 = pounds per cubic foot.
Kilowatts $\times$ 1.34 = horsepower.
Watts $\div$ 746 = horsepower.
Watts $\times$ 0.7373 = foot-pounds per second.
Calorie $\times$ 3.968 = B.t.u.
<i>Cheval vapeur</i> $\times$ 0.9863 = horsepower.
(Centigrade $\times$ 1.8) $\div$ 32 = degrees Fahrenheit.
Gravity Paris = 980.94 centimeters per second = 32.17 feet per second.

TABLE VI.—THE EQUIVALENTS OF OUNCES, PER SQUARE INCH, IN INCHES OF HEIGHT OF COLUMNS OF WATER AND MERCURY

27.71 inches of water and 2.04 inches of mercury equal 1 pound per square inch at atmospheric pressure and 62°F. temperature. Mercury is 13.58 times as heavy as water.

Ounces	Inches of water	Inches of mercury	Ounces	Inches of water	Inches of mercury
0.146	0.25	0.018	7	12.12	0.892
0.292	0.51	0.037	8	13.85	1.019
0.438	0.76	0.055	9	15.59	1.148
0.584	1.01	0.074	10	17.32	1.275
1	1.73	0.127	11	19.05	1.402
2	3.46	0.255	12	20.78	1.529
3	5.20	0.382	13	22.52	1.658
4	6.93	0.510	14	24.25	1.785
5	8.66	0.637	15	25.98	1.913
6	10.39	0.765	16	27.71	2.036

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